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The Society's Research Program

By PRESIDENT DAVID BEECROFT

IT is a welcome opportunity to have the privilege, the honor, to address you as your president, on account of what our Society has accomplished in its brief history of 15 years, and in view of the character of the times through which we are passing and the problems lying ahead with which we must cope. We have passed milestones on the automotive highway but the future is not so clearly defined as we hoped a few years ago it would be; within 12 months we have been brought face to face almost brutally with situations that have been anticipated by few, if any, in the industry. Our concern is with the immediate as well as the more remote future.

Within the last 12 months we have seen waves of industrial depression follow each other in long successive undulations beginning at the Atlantic and ending at the Pacific. We have not only watched these waves traverse the breadth of our land but have seen them pursue their undulatory course across the Americas to the south, across Australia, Africa, Asia and devastated Europe, until we could not locate a single center on the earth's surface that seemed immune from the depression that we are still in the midst of.

FUNDAMENTALS

We are not yet through liquidation, and are barely on the threshold of the adjustment period that lies ahead. These are days when courage is needed; when the fiber of which men are made is under test and when integrity and self-reliance are necessary attributes.

This situation has suggested the subject of these remarks, namely, education, a more general consideration of fundamentals and the development of greater self-reliance. In no field of human endeavor are these qualities more essential than in engineering, and this is especially true in a department of engineering such as transportation, which is so interwoven with most of the other industries of the world.

In transportation the automotive engineer must play his rôle and it must be a just and equitable one. He must give to the railroad what it can best do; he must give to the motor truck that which it can transport more economically that other means; to the commercial airplane should go that which belongs to it. There must be no wasting of effort by any form of transportation attempting to take unto itself that which can be more economically performed by another medium of transportation. Corporational selfishness may exhibit itself in transportation and for a short time make headway, but in the end the efficient will triumph; in transportation the straight line, representing the greatest efficiency, will win. Recently the different kinds of transportation were warring among themselves, vainly trying for the victor's share of the spoils, but a change of attitude has been exhibiting itself and now railroads are admitting the right of the motor truck in certain divisions of the transportation world. Let the law be, render unto Cæsar that which is Cæsar's and to God that which is God's.

If the engineer is to maintain the status of his profession, and follow the ideal of seeking after the truth, the pursuit of education must be an ever-present consideration. He must ever breathe the spirit expressed in the words of Ulysses, "To follow knowledge like a sinking star beyond the utmost bound of human thought . . . to sail beyond the sunset and the paths of western stars until I die . . . to strive, to seek, to find, and not to yield. . . ."

For the first time in our industry, since we shed our swaddling garments, we are measuring ourselves with the other automotive manufacturing nations. Before the war we were not an exporting nation, as the factory expansion which gave us the production we have today was developed during the early years of the war. Previous to 1914 our foreign trade was a diversion; today it is a necessity. We find ourselves with factories of greatly expanded capacity, greater capacity than our domestic demands can absorb, and are faced with the alternatives of remaining in and increasing our position in the markets of the world or reducing output and depriving ourselves of the lower production costs we attained by virtue of volume produced. For six years we have been in the markets of the world in a more or less representative way but, lest we forget, let it be recalled that the world markets sought our wares and the world's buyers knocked on our doors; our task was easy—we opened the doors, took the remittances and delivered the merchandise. "Easy come, easy go," is an old proverb, but still applicable. We might also further fortify ourselves by borrowing from Divine Writ words perhaps suitable to our foreign trade situation, namely, "Wherefore let him who thinketh he standeth take heed lest he fall."

[1] Presidential address delivered at the Semi-Annual Meeting of the Society, West Baden, Ind., May 24, 1921.

Perhaps to some engineering and foreign trade may seem far afield, but only last week the president of one of our largest railroad systems in his testimony before the Interstate Commerce Commission attributed the present decline in railroad traffic to the falling off of our foreign trade, which decreased $657,000,000 in three months of 1920 as compared with corresponding months in 1919. Not only has foreign trade affected our railroad systems but it is affecting all industries, and whatever factors affect the industries with which he is connected should properly be a matter of concern to the engineer.

Foreign trade, while directly a merchandizing activity, has a very close connection with engineering, and its future is dependent to no small extent on the knowledge the engineer shall have of the conditions and the character of the people in the countries in which his product must perform. Some four years ago when our export sales departments ruled that the magneto was not necessary equipment because of our developed battery ignition-systems, it was soon learned that conditions in different countries demanded the magneto and those companies that fitted the magneto were well repaid. Other companies which for a time resisted were soon converted. In the end it was an engineering question; while from a merchandizing standpoint the dropping of the magneto seemed desirable in that it facilitated production, yet in long-distance view, which should be the engineering view, the magneto proved desirable. It is because of many situations of this kind that engineering must be more intimately associated with such factory activities as merchandizing and maintenance as well as design and production. The factory engineer becomes immediately a direct party to foreign trade when sales in certain lands are decelerated due to gasoline selling at $1 per gal., or in another country where the tax is $5 per year per hp. Here we get a closeup realization of how interwoven engineering is with merchandizing, and how only the engineer thoroughly familiar with the fields to which his product goes can serve his company best.

The Engineering and the Sales Departments

The engineer has, moreover, not sufficiently weighed the factors involved in domestic trade. Two years ago we had no world competition but today with European nations entering the field and some of them having been actively in it for a year, it is necessary, if we are to retain our position, to give early consideration to this and related questions. To draw a parallel from our domestic field, two of our large farm tractor manufacturers have within the last 30 days concluded that the high cost of fuel in comparison with animal power has been one of the deterrents in the sale of farm tractors. If this becomes a considerable factor at home where fuel is cheap and the machine is marketed at low cost as compared with prices at which it must be sold in foreign fields, how much more of a consideration does it become with $1-per-gal. fuel and machine prices nearly doubled due to freight, insurance, customs and shipping charges? Only one year ago we found serious engineering defects in certain cars in a section of our own country, due to physical characteristics on the Pacific coast. The engineer was ignorant of the facts and, still worse, obstinate in his error. It was the sales end that finally converted the engineer, whereas it is the duty of the engineer to be a guide to the sales department and correct it when necessary. The engineering mind must give consideration to those factors that directly affect his company.

Let us be constantly mindful of the fact that automotive engineering is far from completion. While in the last 30 years, scarcely a generation, we have seen a measure of progress, hardly comprehended today, the goal lies far beyond the distant hills. Education is the greatest need in present-day engineering. In pursuing education along its devious paths we can gain hope and inspiration from Gotthold Ephraim Lessing, the German philosopher who in his Education of the Human Race, in referring to the broad subject of evolution, said, "Go thine inscrutable way, Eternal Providence, only let me not despair in Thee because of this inscrutableness. Let me not despair in Thee even if Thy steps appear to be going back. Is it not true that the shortest line is always straight?"

Education should only begin when college doors close behind us. The man does not truly live who concludes at any time in his life that his education is completed. "Education gives to man nothing he might not educe for himself. It gives him that which he might educe for himself, only easier and quicker. This in the same way that revelation gives nothing to the human species which the human reason left to itself might not attain, only it has given and still gives the important of these things earlier."

Research

The extension of knowledge and securing it more quickly and easily is one of the functions of the Society that your Council has given consideration to since the first of the year and was under consideration last year. In connection with the term research as related to the Society, the dominant thought is the securing of certain knowledge earlier and with less expenditure of effort and with greater conservation of the talent available. Your Council has seen fit, after deliberate and mature study, to push actively the creation of a Research Department that will take its place in the Society activities along with the standardization work. It is impossible to see what development may be ahead and what expenditures this department may require in the next few years, but in magnitude the research organization should exceed that of standardization, and the growth of the department is dependent only on the support it shall receive from the membership.

Let us analyze briefly what the research is as contemplated and what it would mean to the Society and the members:

First, there has been no thought of creating a special research laboratory for the Society in which to carry on experiments necessary in any research; but rather that existing laboratories in the industry and outside of it shall be utilized for such work. The finances of the Society would not permit of creating a special laboratory, and the feasibility of this would be gravely questioned at this time. There is in Government bureaus, such as the Bureau of Standards and the Bureau of Mines, in our college laboratories and in our industrial laboratories, ample equipment for all necessary research, and it would be wasteful to neglect the intelligent use of this.

Second, there has been no thought in the research program of encroaching on what might be termed the secret developments of corporations. There is nothing communistic or flavoring of engineering socialism in the plan. The research program contemplates nothing more than what has been accepted as beneficial cooperation in association of manufacturers. We have for years, even for centuries, considered the industrial associations formerly known as guilds not only desirable but essential for the efficient promotion of trade. We are today accom-

plishing through them results not considered within the pale of possibility a few years ago.

There is nothing in the research planned that usurps programs of the engineer today. There is no thought of extracting the scientific discoveries of one company and distributing them broadcast to the remainder of the industry. The thought is to do the things that have thus far been left undone; to obtain certain knowledge not only more easily but more quickly, and to be more certain of its accuracy.

For our present convenience, we may consider research under three heads or classes:

There is what might be designated explorational research in different fields. This is pure scientific research, largely conducted by individual scientists with the thought of extending the boundaries of human knowledge. It is akin to the work of an explorer starting on a mission of discovery in an unknown field, a case in point being that of the Scandinavian astronomer who has lived in his observation station within the Arctic Circle, where for 20 years he has studied the Aurora Borealis and as a result established certain scientific relationships previously unknown and by virtue of which new ideas so to speak have been brought into the field for intensive study by the scientific and engineering world. With this class of work the Society has little to do. It has not been considered to be one of our major activities. It must be left to the physicist, the mathematician and the general scientist.

Closely related to explorational research is what might be designated intensive research in the fields brought within the ken of engineering by explorational work. Our program should be closely connected with this. It is here that economy of effort is most needed. This intensive research involves cooperation and organization, and the talent used in it can rarely be used in explorational research. An example of research of this kind is that relating to fatigue in metals. This involves an almost endless number of experiments and tests extending over long periods of time and covering a wide range of materials and conditions. Some of our universities have been conducting researches of this character in which tests have been carried on for over a year and are still far from completion. These experiments do not per se give the answer to fatigue but the findings must be interpreted and from the interpretations deductions made so that it will be possible to predict the fatigue resistance of metals. Intensive researches of this and similar character are supported liberally by large industrial corporations which have their own research laboratories and recognize the magnitude and the urgent necessity of the work.

The third kind of research is industrial or development research which for convenience may be designated competitive research among manufacturers in the same field of industry. It is research in which our program has no part. Competitive or development research has to do largely with the design of a part or completed unit or entire vehicle for manufacture or patent protection. This is the research with which many corporations are directly concerned and upon which their laboratories are largely engaged. It is generally carried on behind locked doors on which is the sign, "Positively No Admission." It is the research which provides the manufacturing secrets of a corporation. Our contemplated program leaves this where it is today, solely with the manufacturer where it rightly belongs. Our program does not even hint at unlocking the door or removing the sign. Competitive or development research is and must remain a company activity.

Careful discrimination between this competitive or development research and intensive and explorational research should be kept constantly in mind. Research of the intensive character will tend to increase rather than restrict the possibilities for competitive and development research. As an example, considering the influence of turbulence in connection with the internal-combustion engine, the knowledge of turbulence elevates the standard of practice of the entire industry to a higher and wider level on which each corporation interested has greater possibilities for individual development, patent protection, and the like. It is only by a recognition of this that we will advance in knowledge as the demands of the day require.

Necessity of Cooperation

Research gives to man nothing he might not educe for himself, but it gives it to him more easily and quickly. So with intensive research, the cost is so great and the time needed so long that only by cooperation and organization in the long chain of tests can we secure the knowledge more quickly, easily and cheaply and be able to incorporate the results in industry sooner. Do not forget, art is long and time is fleeting.

One great need today is the training of men competent to carry on the intensive research activities of not only our colleges but of our industries. Our colleges are not graduating enough men of research caliber to meet the situation unless methods of cooperation and organization are used. We must conserve the human material we have available. No persons are more conscious of the shortage of research personnel than our college heads and no one regrets it more than they do. Too frequently the curricula are not conducive to the proper result.

Up to the last few years there has been little thought of cooperation for conservation in college research, and still less thought of cooperation in intensive research in industrial plants. There is today a lack of appreciation of research by not only the engineers but by company executives, and just as our standardization program met with opposition at its inception and still meets with opposition in some quarters, so we can anticipate opposition to our research work until it is more adequately understood. The work will not deprive any engineer of his present work, but enable him to obtain results which otherwise might not be possible. It is going to provide the engineer with an arsenal well stocked with knowledge that will serve him and speed his efforts to better work.

Our Research Committee has had one specific objective in mind, namely, to secure as the director of the research work a man not only competent to carry on intensive research but capable of the more difficult task of correctly interpreting results obtained from such research. There is a lack of uniformity of interpretation of results obtained, with consequent failure to reach conclusions and make correct deductions. In some intensive researches conducted by one engineer certain factors are neglected, so that comparison of results of experimentation covering the same field by other groups of engineers cannot be made. The measure of result that should be achieved from the effort expended is not achieved. The money has been expended; the useful time has been consumed; the human energy has been consumed, but the results are not what they should be. A competent director should, with reasonable cooperation, be able to eliminate such losses. Had we a

surplus of human talent our program for intensive research would scarcely be necessary but with a shortage of available men the need for such cooperative work becomes imperative. In considering our research problem we suggest that the word research be divested of early associations, that your conception of it be revised, and that it be weighed in the light of the work to be done, the end to be accomplished and the tools and personnel available, keeping in mind the differences between intensive and competitive research. Do not forget that the confines of competitive and development research are not to be invaded or molested by our work but that the field for this is to be broadened and lengthened by the intensive-research program contemplated. No time could be more appropriate than the present for the commencement of such work. In times of prosperity, when a company wishes a result along a certain research line, the answer is wanted almost immediately. This is not possible. By beginning now the program will develop with a readjustment of the industry, and some progress will have been made by the time urgent demands are coming in.

There is no thought, in the cooperative intensive-research program outlined, of interfering with or discouraging similar researches by corporations suitably equipped and manned. In the case of similar researches in the electrical and other fields those corporations privately equipped for research have been liberal in contributing their aid and finances for general research. This has a very desirable influence by creating self-reliance on the part of the individual engineer. Those institutions that have made greatest progress in such work have had largely to train their own personnel. Colleges by virtue of their curricula do not make research engineers. Individual training is necessary. Self-reliance is essential. Ability to proceed step-by-step from the known to the unknown is requisite. The French philosopher René Descartes of the sixteenth century gave us a good example of what can be accomplished when self-reliance is developed. In his Discourse on Method in which he laid the foundation work for modern thought and made possible the whole modern philosophic development, he says, "Men should gather the greatest satisfaction from progress made in the search after truth. In the same way I thought that the sciences contained in books, composed as they are of the opinions of many different individuals massed together, are further removed from truth than the simple inferences which a man of good sense using his natural and unprejudiced judgment draws respecting the matters of his experience. . . . The long chain of simple and easy reasonings by which geometers are accustomed to reach the conclusions of their most difficult demonstrations has led me to imagine that all things, to the knowledge of which man is competent, are mutually connected in the same way, and that there is nothing so far removed from us as to be beyond our reach, or so hidden that we cannot discover it, provided only we abstain from accepting the false for the true, and always preserve in our own thoughts the order necessary for the deduction of one truth from another."

Today, as always, the greatest problem in engineering is the engineer. Man dominates here as in all other spheres and only in proportion as the man, tagged as he may be, with the title of engineer, etc., plays his part and considers the attainment of knowledge as the chief end in life, will our industry develop and we reach our full stature and be capable of measuring ourselves successfully with the nations of the world. We have been parties to an industry whose lot has been an easy one. As an industry we have grown by leaps and bounds. Our progress has been comparable with that of an army in the field that has made great gains of territory without adequately consolidating its position as it progressed and suddenly finds itself in an unexpected situation. We have not at all times taken accurate measure of our progress nor taken a sufficiently long-distance survey of the future. We are now at a time when looking ahead is more needed than it has been heretofore. We require better qualifications to cope with the situation at present than were necessary in the past.

EXTENT AND EFFECT OF S. A. E. STANDARDS

WITH the adoption by the Society of the recommendations approved at the May Standards Committee Meeting, the total number of S.A.E. Standards and Recommended Practices will reach 250. The importance of the work represented by these standards can be appreciated more fully by the following analysis of the 224 standards adopted up to April, 1921. If automotive parts, fittings and materials were fabricated in accordance with all the dimensions and analyses specified in the 224 standards, the total number would be 2307. Many of the parts and fittings standards, however, specify dimensions for several parts. For instance, the S.A.E. Standard for Steering-Wheel Hubs specifies dimensions for the steering-column tube, the steering-wheel hub and the steering-wheel-hub nut, three separate parts. If all the parts and fittings were separated into their components, the total number of parts and materials would be 3482.

The accompanying table indicates the manner in which the standards are divided.

ANALYSIS OF S. A. E. STANDARDS

Standards	Number of Standards	Number of Sizes and Analyses	Total Number of Parts and Analyses
Powerplant	23	186	447
Electrical Equipment	38	153	204
Parts and Fittings	64	975	1,675
Materials	24	361	368
Transmission	13	225	225
Axle and Wheel	6	45	106
Tire and Rim	15	134	186
Frame and Spring	15	169	185
Control	9	24	43
General	17	35	43
Total	224	2,307	3,482

It will be noticed that the number of standards adopted for parts and fittings is almost twice as large as for any other group, and that the total number of parts is about four times as large as for any other group. This is because the standardization of parts and fittings affects individual design the least and also because they are, as a rule, commodities sold in the open market.

An analysis of the data showing current practice obtaining at the time of standardizing several parts and materials indicates that the reduction in sizes or analyses averages about 40 per standard and that the ratio of reduction of sizes in current practice to sizes adopted as standard is more than 6 to 1. It follows apparently that standardization of parts and materials reduces the number of sizes and analyses employed in regular production 80 per cent.

Developing a High-Compression Automotive Engine

By FRED C. ZIESENHEIM[1]

SEMI-ANNUAL MEETING PAPER *Illustrated with* CHARTS AND PHOTOGRAPHS

TO facilitate the logical consideration of the development of a high-compression engine for automotive purposes, this paper is divided into parts (*a*) the fuel problem of the automotive industry, (*b*) the selection of the most economical internal-combustion engine for adaptation to automotive purposes and (*c*) the details of the development work undertaken.

THE FUEL PROBLEM

The progress of our civilization has been synonymous with the increased use of mechanical power. The sources of energy whence this power is derived are water, coal and petroleum. Of these, petroleum is the most easily produced, transported and converted into power. The conversion of petroleum into power is accomplished most easily and economically by the internal-combustion engine. The importance of the internal-combustion engine can hardly be realized, so recently has it been developed and put to work on the tasks of our civilization. It has become the means of annihilating space on land, in the air and on the sea, and of transportation for more people than any other single agency. Deserts have been watered and made to produce food and the earth now produces food free of toll by animals in return for their energy. At present about 75 per cent of the world's mechanical power is being produced by the internal-combustion engine.

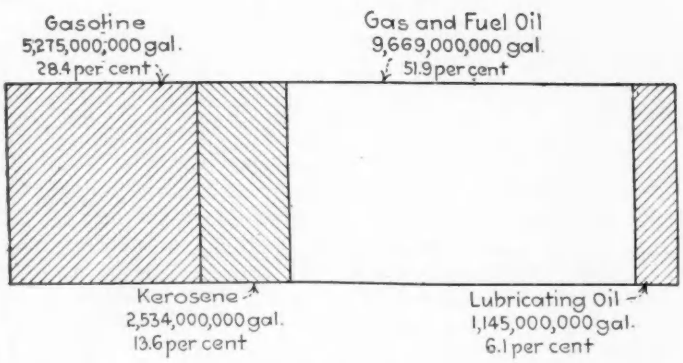

FIG. 1—ESTIMATED PRODUCTION OF PETROLEUM PRODUCTS IN THE UNITED STATES FOR 1920

The estimates of geologists and of oil producers as to the petroleum resources of the world show a considerable variance. Even accepting the optimistic view of the oil producers as to the supply available for the future, the necessity for the immediate conservation of this valuable natural resource is strikingly evident. The important products of crude petroleum are gasoline, kerosene and gas, fuel and lubricating oils. The estimated production of the United States for 1920 is given in Fig. 1 and totals 18,623,000,000 gal., an increase of 17.4 per cent over that of 1919. Mexico produced 6,711,600,000 gal. in 1920, an increase of 83.5 per cent over that of the previous year.

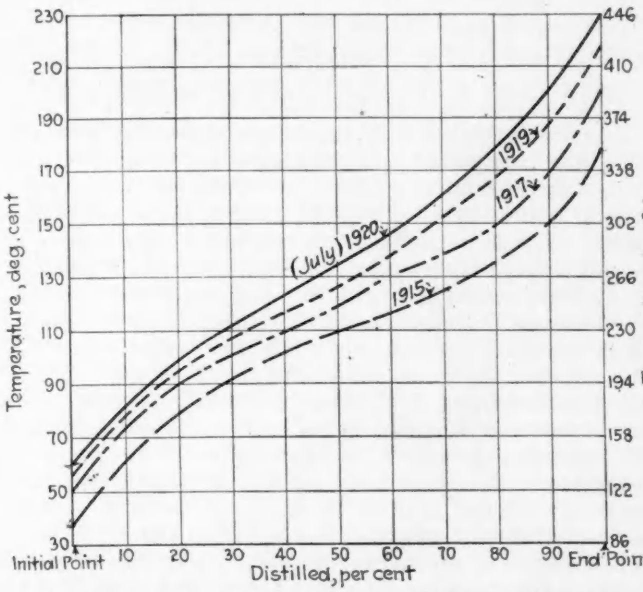

FIG. 2—TREND OF THE CHANGE IN THE VOLATILITY OF GASOLINE FROM 1915 TO 1920 SHOWING THE RISE IN THE END-POINT

The production of crude petroleum, although increasing, is not keeping pace with the consumption of engine fuel. The production of crude petroleum has increased

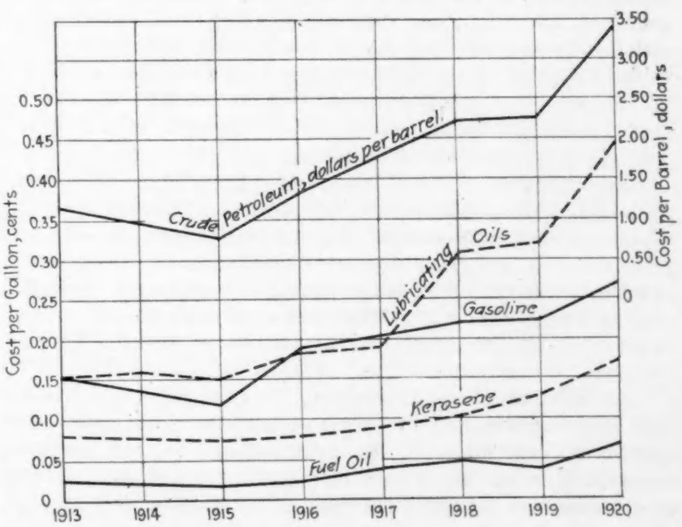

FIG. 3—AVERAGE PRICE OF CRUDE PETROLEUM AND ITS PRINCIPAL PRODUCTS FROM 1913 TO 1920

[1]M. S. A. E.—Development engineer on high-compression engines, Carnegie Institute of Technology, Pittsburgh.

nearly 100 per cent in the past 10 years; whereas the production of gasoline has increased nearly 600 per cent in the same period. The deficit is being met by constantly lowering the volatility of the fuel supplied. The change in volatility over the period from 1915 to 1920 is shown in Fig. 2, from data of the United States Bureau of Mines. If it is necessary to make additional cuts into the less volatile portions of the crude oil, engine fuel will rapidly become of such a nature that the majority of present-day automotive engines cannot be operated upon it. Ninety per cent of the world is still lighted with kerosene lamps; therefore, the market for kerosene, firmly established previous to the development of the internal-combustion engine, will dispute the priority of the automotive industry to that portion of petroleum products. Fig. 3 shows the average price of crude petroleum and its principal products from 1913 to 1920.

Methods of Increasing the Engine-Fuel Supply

Gasoline substitutes from other sources than petroleum have been suggested, but the quantity of engine fuel required is so enormous that the development of a substitute in the requisite amount must be a matter of many years. The only products of immediate significance are benzol, a by-product in the distillation of coal, and alcohol, made from organic materials. The present total output of benzol is only 2.5 per cent of the country's gasoline requirement. The output, while increasing, is not increasing as rapidly as the consumption of gasoline. By extreme measures, including by-product recovery from all the coal coked, stripping all city gas supplies for light oil and other increases to be expected in the next few years along these lines, we cannot anticipate that much over 5 to 10 per cent of the gasoline demand of today could be met with benzol. The material available for the manufacture of alcohol is unlimited, but the volume of the material required, the investment necessary and the heat required for producing alcohol in quantity, constitute a serious handicap to large-scale production. The present production of alcohol is about the same as that of benzol. The production of alcohol in the tropics has been suggested, but authorities state that it would require 5 acres of tropical vegetation under the most favorable conditions to produce enough fuel to supply a motor truck for 1 year. It is highly improbable that alcohol or benzol can be produced in sufficient quantity in the immediate future to become substitutes for gasoline but, when added to the gasoline supply, they may assist in meeting the situation. Mixing alcohol with gasoline requires the use of a binder to render the two liquids miscible. Benzol is satisfactory for this purpose, about 10 to 15 per cent being required. The amount of alcohol that can be produced for addition to gasoline is limited, therefore, by the supply of benzol available.

The oil shales of the Rocky Mountains, Texas and other districts, hold promise for the future, although the cost of production may exceed that of the present petroleum products. A program for the more economical and efficient utilization of present petroleum resources will be applicable likewise to the shale-oil resources which appear to be the most logical source of supply for the future.

The only commercial method for procuring an engine fuel suitable for use in gasoline engines from kerosene, gas oil or fuel oil is by the pressure-distillation method, commonly called the "cracking" method. Cracking is the most important factor capable of enlarging the supply of gasoline from a given supply of crude petroleum. While cracking processes are legion, very few have attained a working efficiency greater than 40 per cent, thereby entailing a loss of 60 per cent in the process.

The utilization of the fuel-oil portion can be accomplished in two ways. One is to convert it into gasoline through the cracking process, which entails a loss of 60 per cent; the other is to develop an engine capable of operating on fuel oil. To quote J. O. Lewis, chief petroleum technologist of the United States Bureau of Mines:

> Cracking heavy oil into gasoline is an economic loss which should be tolerated only until the problem of a satisfactory automotive engine for consuming the heavy fuels can be solved. In cracking there is both a loss of material and a loss because of manufacturing costs; yet the gasoline resulting yields hardly half the power in the automotive engine of today that the original oil would in a Diesel type of engine.

Commercial cracking processes to date have been able to convert into gasoline only kerosene, gas oil and light fuel oil, leaving the major portion of fuel oil untouched. The gasoline produced by the cracking process is unstable. In time tars are formed which separate out and cause trouble in engine operation.

Characteristics of Present Engine Fuels

Considerable difficulty is being experienced at present in operating automotive engines on the class of fuel supplied. As the volatility of the engine fuel has been lowered, it has become necessary to heat the charge of fuel and air to induce vaporization. Heating the mixture, expanding it, decreases the weight of mixture that the engine can draw into the cylinders, thereby making the engines less efficient. Due to the difficulty of vaporizing the mixture, it will be composed of a liquid portion and a vaporized portion. The liquid portion of the fuel tends to flow along the walls of the intake manifolds, thereby introducing difficulties in the distribution of a uniform mixture to the various cylinders. Some of the liquid portion will not be burned, but drain past the pistons into the crankcase, and cause dilution of the lubricating oil. If injury to the bearing surfaces is to be prevented, the lubricating oil must be drained frequently and replaced with new oil, thereby adding the loss of the lubricating oil to the economic loss of the fuel.

Another characteristic of the present engine fuel is the detonation which occurs unless the compression is kept at a very low figure. Detonation is the term applied to the combustion knock that results with a particular fuel, when its dissociation limit of pressure and temperature are exceeded. The fuel-and-air mixture disintegrates or dissociates, instead of burning freely and quietly, which allows the hydrogen to combine with oxygen and releases free carbon. The almost instantaneous combustion of the hydrogen produces a pressure wave which travels through the gases, impinges on the combustion-chamber walls and produces the familiar knocking or "pinking" called detonation. Detonation may be merely annoying, or destructive, depending on the compression pressure of the engine. The maximum compression pressure allowable must be kept below the point at which detonation occurs.

The conditions which influence the design of gasoline engines, such as the heat required for vaporization, detonation and dilution of the lubricating oil, are present to a greater extent with kerosene as a fuel than with the slightly more volatile gasoline. Kerosene has a higher heating value than gasoline. It is cheaper per gallon and therefore per heat unit but, due to the condi-

tions enumerated, kerosene engines are necessarily designed to operate less economically than gasoline engines.

The brake thermal efficiency of modern automotive engines, exclusive of aircraft engines, which is the percentage of the total heat energy available in the fuel converted into useful work, will be less than 15 per cent under the most favorable conditions of full load and full speed. The limitations imposed by the modern fuel, such as heating of the fuel-and-air mixture and the low compression necessary for obviating detonation, have a tendency to make the thermal efficiency even less than the figure given. An automobile engine under normal running conditions is delivering less than one-third of its full-load power and, due to its method of control by throttling the fuel-and-air mixture, the thermal efficiency will be less than 5 per cent. The important fact to be realized is that the average automobile engine converts but 5 per cent of the energy of the fuel into useful work under the usual running conditions.

Summarizing the fuel problem of the automotive industry, the demand for gasoline has been and is increasing at a greater rate than the production of crude petroleum, which necessitates the lowering of engine-fuel quality by the addition of less volatile portions of the crude. The present engine fuel is unsatisfactory in that it requires heating for vaporization, causes dilution of the lubricating oil and produces detonation during combustion. These are contributory causes for the low thermal efficiency and the operating difficulties of existing automotive engines. The production of gasoline substitutes is insignificant in comparison with the quantity required. If supplementary sources of non-petroleum origin can be developed that are capable of supplying the requisite quantity, it is highly improbable that the product will be sufficiently volatile for the existing types of automotive engine. The conclusions are that conditions in the petroleum industry, the engine conditions imposed by the present engine fuels and the low thermal efficiencies and operating difficulties of modern automotive engines all indicate the advisability of developing another type of engine capable of generating power more economically from less volatile fuels.

Selecting and Adapting the Most Economical Engine

The internal-combustion engine being admittedly superior to other classes of heat engine in the economical generation of power, a survey of the existing types of internal-combustion engine is necessary to select a more economical type of prime-mover for adaptation to automotive purposes. The selection of a type of engine for a particular class of work involves establishing a common basis of comparison. The prime considerations are ability to perform the work and the cost per unit of work.

A device for the accomplishment of a specific task is superior to another capable of doing the same work, when its overall cost of operation is less. The overall cost of operation includes all items chargeable to the device in the fulfillment of its function. These are divided into capital and operating charges, the former being composed of initial, installation, interest, depreciation and insurance costs; and the latter including fuel, lubricating oil, maintenance, attendance and repair expense. Other considerations are ease of operation and maintenance, continuous reliability, the weight of the engine and the fuel, the saving in the space required for the engine and the fuel, safety in operation, cleanliness and the ability to operate on various classes of fuel with a minimum of changes or adjustments. A comparison of the engineering features of the respective types will assist in establishing their relative merits in point of ability and give an indication of their relative costs.

Classification of Internal-Combustion Engines

Internal-combustion engines can be classified, in accordance with the cylinder compression used, as being of low, medium or high compression. The present type of automotive engine is a low-compression engine in which a mixture of fuel and air is drawn from a carbureter into the working cylinder. The charge is compressed to about 40 to 60 lb. per sq. in. and then ignited by an electric spark. This type of engine is apparently incapable of further development to securing greater fuel economy because of limitations imposed by the fuel, as previously stated. Some European developments of the low-compression engine have abolished the curbureter. A French system, the Bellem, injects the kerosene fuel into the combustion-chamber during the end of the compression stroke. At a recent French fuel competition this system, fitted on a Unic five-passenger car weighing 4386 lb. and having an engine of 4-in. bore and 5.9-in. stroke, gave a consumption rate of 17.9 miles per gal. of kerosene. An English engine, the Blackstone, has a low compression of from 80 to 130 lb. per sq. in. and injects kerosene fuel by a puff of compressed air into the combustion-chamber during the end of the compression stroke. At the British tractor trials last year a track-laying Blackstone tractor with a three-cylinder 25-hp. engine had, in the heavy plowing tests, the lowest fuel consumption per acre plowed. In the light-plowing test it was among the first five. However, the development of a kerosene engine offers no appreciable relief from the fuel situation, because the supply of kerosene is even more limited than that of gasoline. An engine developed to use a lower grade of fuel than kerosene, will usually operate successfully on kerosene.

The compression of engines included in the medium-compression class ranges from 100 to 375 lb. per sq. in. These are designated "semi-Diesel," "surface-ignition" or "hot-bulb" engines; all three terms being applied to the same type of engine. The operation of a medium-compression engine is that pure air is compressed in the cylinder and the oil fuel, injected into the combustion-chamber at a definite point in the compression stroke, is usually projected against a hot surface comprising an uncooled portion of the combustion-chamber. The volatile portions of the sprayed fuel are gasified and ignited upon striking the hot surface. The heat supplied jointly by the compression, the hot surface and the early ignition contributes to the vaporization, gasification and ignition of the charge. The combustion is at constant volume, being initiated as an instantaneous explosion, although it continues throughout the working stroke as after-burning, depending upon the characteristics of the engine, the fuel and the load. The pressure-temperature conditions existing when the fuel impinges on the hot ignition surface are such that dissociation occurs immediately, releasing free carbon and the hydrogen which produces the marked detonation of this type of engine. Detonation, which becomes objectionable in low-compression engines with the use of heavier fuels than first-run gasoline, is an inherent characteristic of all medium-compression engines. The detonation can be limited by control of the combustion-chamber temperatures; this is usually effected by water injection. The water, conveyed by the incoming air into the combustion-chamber, must be of sufficient quantity so that the heat required for its vaporiza-

tion will lower the combustion-chamber temperature to the desired maximum and thereby limit the dissociation and detonation that can occur. The heat absorbed by the water is a loss, inasmuch as it is not converted into useful work but is discharged with the exhaust. Medium-compression engines, not equipped with water injection or other satisfactory means of temperature control, are usually given a 20 per cent lower power-rating than engines of similar size that are so equipped. It is often necessary to dispense with the water injection, because the water available contains sulphur, alkali or other elements that are injurious to the cylinder walls.

The advantages of the medium-compression engine as compared with the low-compression type are that

(1) It operates on light fuel oil, distillate, gas oil or kerosene; fuels that are cheaper than gasoline
(2) The fuel consumption is slightly lower at full load and considerably lower at partial loads
(3) It is a simpler engine. The design is usually a two-stroke-cycle, valveless, requiring fewer parts
(4) It has lower operating costs, due to its simplicity and its lower consumption of cheaper fuels

As compared with the high-compression engine, the medium-compression engine is simpler and lighter, and has lower capital and operating costs, in addition to operating on lower compression pressure. The disadvantages of the medium-compression as compared with the low-compression engine are that

(1) It operates on a higher compression pressure
(2) The hot bulb or ignition surface must be heated before starting the engine
(3) It requires water-injection or other means for controlling ignition temperatures
(4) There is difficulty in controlling and maintaining the temperature of the ignition surface for varying conditions of load, for different grades of fuel and for different conditions of cooling and injection water
(5) It has the marked detonation characteristic of all medium-compression surface-ignition engines
(6) Operating troubles result from the nature of the combustion process; that is, dissociation with the release of free carbon, the fouling of the combustion-chamber with carbon and the necessity of cleaning it to prevent preignition. The exhaust is usually black
(7) It has poor speed-regulation with varying load conditions, which is especially objectionable when driving electric generators
(8) High maximum pressure results from early fuel injection

As compared with the high-compression engine, the medium-compression engine has all of the disadvantages in the foregoing list except (7); a higher fuel consumption; an inability to use as low a grade of fuel and therefore a higher operating cost; an inability to use various grades of fuel without major adjustments in the combustion system; and dilution of the lubricating oil.

The principal disadvantages of medium-compression engines are the necessity for preheating the ignition surface before starting; the difficulty in maintaining and controlling the ignition temperatures at all loads; the enormous stresses placed on the mechanism due to the excessively high explosion pressures which result from the necessarily early injection of the fuel; and its inability to operate on other than a narrow range of fuel without extensive changes in the combustion system. These disadvantages are fundamentally inherent in this type of engine and are such as to eliminate the medium-compression engine from consideration as a development for automotive purposes.

The high-compression oil engine produces power more economically than any other known type of heat engine. For comparison with a steam engine using fuel oil, George Otis Smith, director of the United States Geological Survey, states that

> The very facts that support the argument for the marine use of fuel oil, the greater efficiency and economy of space and labor, can be cited in favor of the internal-combustion engine of the Diesel type against the steam engine. The increased thermal efficiency of the new engine, with its resulting addition to available cargo space or to cruising radius, is more than 2½ times that of the steam engine. The experience of the Bethlehem Steel Co. is that its new oil-engine ore carrier, the Cubore, in continuous service between Cuba and Sparrows Point, Md., uses only 36.7 per cent of the fuel oil consumed by a sister ship differing only in that it has the most modern type of steam plant. The tremendous economy thus possible in the marine consumption of fuel oil demands the immediate adoption of internal-combustion engines if the world wants to make the largest use of its oil resources for the longest time.

In a paper entitled Working Processes of Internal-Combustion Engines[2] which was presented at the 1919 Semi-Annual Meeting of the Society, Prof. C. A. Norman of the Ohio State University stated that

> An investigation by the United States Department of Agriculture reveals the fact that 68 per cent of all tractor-engine troubles occur in the magneto, spark-plugs and carbureters or in the accessories of the present-day automobile engine. Bearings, cylinders, piston-rings, valves and springs, lubrication and starting systems, that is to say, the parts common to all classes of combustion engines, gives rise to only 32 per cent of the troubles. From the point of reliability alone we have then a most serious reason to be on the lookout for some new type of automotive engine.
>
> The reason from the point of view of fuel utilization is even more compelling. An engine may be considered as having good carburetion that turns into shaft horsepower more than one-fifth of the fuel supplied it. Four-fifths is regularly wasted in our present automotive engines. Yet the fuel must be a particularly high grade one, a liquid meeting severe requirements of volatility, liquidity and the like. The question of a continued supply of such a fuel is becoming a serious one. . . . Actually, Diesel engines with compressions as high as 550 lb. per sq. in. have, on the test stand, reached utilizations of 36 per cent.

In the operation of a high-compression oil engine, pure air is compressed in the engine cylinder to a pressure of from 350 to 500 lb. per sq. in. The temperature attained by the air, due to its compression, is from 950 to 1100 deg. fahr., as is shown in Fig. 4. The oil fuel is injected in a finely divided state into the highly compressed air of the combustion-chamber, commencing just previous to the moment when the piston reaches the top dead-center. The injection is continued for a specific period, depending upon the engine load. Auto-ignition of the injected fuel takes place, due to the temperature of the air compressed within the cylinder. The combustion which ensues is slow burning; the rate of fuel injection is such that the combustion takes places at constant pressure. The combustion of the fuel releases heat and does expansive work on the piston, thereby converting the heat energy of the fuel into mechanical work. The engine cylinder is scavenged free of combustion

[2] See The Journal, July, 1919, p. 3.

gases and filled with pure air, and the operating cycle is then repeated.

The advantages of the high-compression engine as compared with engines of low and medium-compression are that it has

(1) Unsurpassed fuel economy and high thermal efficiency, irrespective of the size of the engine
(2) Ability to maintain its economy and thermal efficiency for fractional as well as full loads
(3) Ability to operate successfully on non-volatile fuels, heavier and cheaper than gasoline or kerosene; in fact, on any hydrocarbon fuel that can be rendered liquid
(4) Freedom from the occurrence of detonation
(5) Complete and clean combustion of the fuel, with a clear exhaust
(6) High torque characteristics at speeds lower than those of normal rating
(7) Easy instantaneous starting, with a rapid assumption of full load
(8) Excellent speed regulation throughout its speed and load range
(9) A greater possible simplicity of design
(10) No water-injection or other devices for the control of combustion-chamber temperature
(11) Absence of the fuel-distribution problem of the low-compression engine
(12) No lubricating-oil dilution as in both the low and the medium-compression engines
(13) Ease and simplicity of operation
(14) Reliability and low operating costs
(15) Increased safety in operation from the use of a less volatile fuel of high flash-point

As compared with low and medium-compression engines, the disadvantages of the high compression engine are the

(1) Heavy construction, necessitated by high compression
(2) Difficulty of maintaining high compression under long-sustained service
(3) Requirement of higher capital costs for the heavy construction
(4) Requirement of a higher initial cost necessitated by the excellence of the materials and workmanship
(5) Higher initial cost resulting from the greater ability and amount of work required in the development of the engine.
(6) Educational work necessary for the operation and servicing of a new type of engine

The steps in the complete combustion of a liquid fuel in a high-compression internal-combustion engine are the pulverization of the liquid fuel into fine particles; the vaporization or gasification of the fuel particles; the intermixture of the gasified fuel and oxygen; the decomposition and recombination of the fuel gases with oxygen; and the combustion with an attendant release of heat.

Pulverization, being the initial step, is the predominant one in achieving complete combustion. The manner and effectiveness of the other steps in fulfilling their functions are determined by the extent to which atomization is achieved by the pulverization. The conditions for optimum pulverization are (*a*) an instantaneous pulverization of the fuel upon its initial discharge into the combustion-chamber, with no dribble at the beginning of the discharge; (*b*) a maximum pulverization of the fuel, the ideal condition being complete atomization and that the fuel be so finely divided that the particles are without finite mass; (*c*) uniformity of pulverization throughout the period of discharge irrespective of the rate of discharge, which may vary within narrow limits; and (*d*) uniform maximum pulverization of the fuel up to the moment of cessation of discharge, with a clean complete cut-off of the fuel and no after-dripping. High-compression engines can then be compared on the basis of the means and the extent to which pulverization and consequently combustion are achieved.

High-compression engines are classified, as to their method of injecting the fuel into the combustion-chamber, into the three general classes of air, gas-pressure and mechanical injection.

Air Injection

In the air-injection method the fuel is injected into the combustion-chamber by highly compressed air at a pressure exceeding that of the cylinder by from 200 to 600 lb. per sq. in. The Diesel engine, which uses air-injection of the fuel, is the original high-compression engine and has undergone a consistent development during the 28 years since its invention by the German engineer, Dr. Rudolph Diesel. The Diesel engine has been adapted to practically every class of prime-mover. It has been built in sizes of cylinder of from 2 to 2000 hp. The largest marine engine totals about 6000 hp.

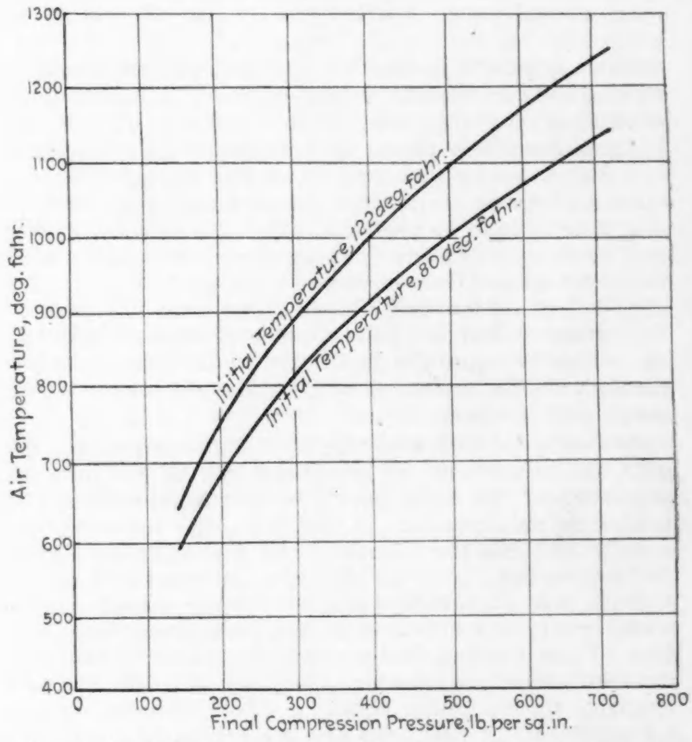

Fig. 4—Air Temperatures for Different Final Compression Pressures

As shown in Fig. 5, the Diesel engine requires a multistage air compressor for delivering the injection air at a pressure of 800 to 1200 lb. per sq. in. Fig. 6 shows a sectional view of a Diesel-engine fuel-injection valve. The interior of the fuel-valve, being in communication with the air-compressor, is filled with air at a pressure of from 800 to 1200 lb. per sq. in. The oil fuel, pumped against the air pressure existing in the injection-valve, is delivered into the lower portion of the chamber surrounding the valve-stem. The quantity of fuel delivered is regulated by a governor to conform with the engine load. When the valve-stem is lifted by a cam acting through levers, the air forces the fuel through the pulverizer into the combustion-chamber. The pulverizer

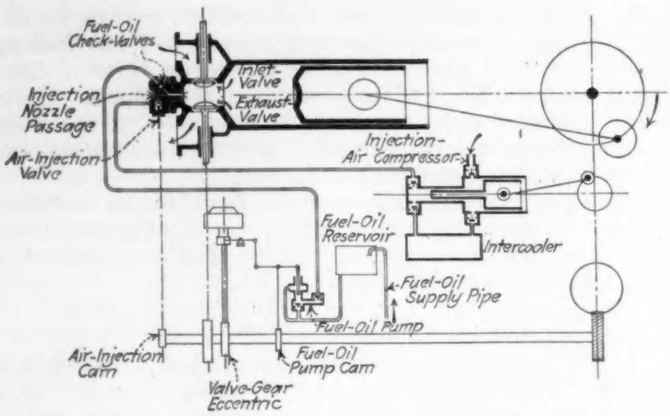

FIG. 5—DIAGRAMMATIC ARRANGEMENT OF THE DIESEL ENGINE MECHANISM

forms a labyrinth passage for distributing and intimately mixing the fuel with the injection air. The injection air, expanding from its pressure of from 800 to 1200 lb. per sq. in. to the 500 lb. per sq. in. pressure of the combustion-chamber, breaks up the fuel into a fine spray. The combustion-chamber air is at a temperature of about 1100 deg. fahr., due to its pressure, which causes auto-ignition and combustion of the fuel particles. The combustion releases heat and does expansive work on the piston. The distribution of the fuel charge throughout the combustion-chamber and the intermixture of the gasified fuel and oxygen are greatly facilitated in the Diesel engines through the turbulence created by the discharge and expansion of the injection air.

As compared with other types of high-compression engine, the advantages of the Diesel engine are that the expansion of the injection air is an effective method of achieving pulverization of the fuel; the injection air, usually 25 times the volume of the fuel, supplies oxygen for combustion; and the injection air produces a turbulence, not duplicatable easily by other known means, which materially aids in the distribution and intermixture of the gasified fuel and oxygen. Due to the long development of the Diesel engine, sufficient data are now available for designing injection valves to meet particular conditions of fuel, load and other requirements, provided these conditions do not differ materially from those already met in practice. The influence is now known of

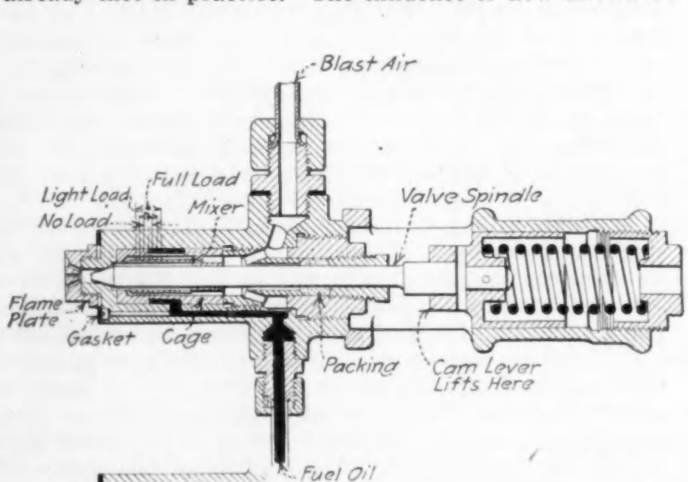

FIG. 6—SECTIONAL VIEW OF A DIESEL-ENGINE FUEL-INJECTION VALVE

pulverizers, valve-lift and orifices, upon atomization, the rate of combustion and the form of the indicator diagram. The disadvantages of the Diesel engine, as compared with the other types of high-compression engine, consist of the complication of the air-compressor which generates the high-pressure injection-air; the higher initial costs and capital charges necessitated by the air-compressor; and the operating difficulties incident to the use of the air-compressor as the majority of Diesel-engine troubles are attributable to the air-compressor and its accessories. Operating difficulties result in lessened reliability and are a source of operating expense.

The air-compressor comprises the usual mechanical parts such as pistons, connecting-rods and valves. The compressor must be of not less than three stages and proportioned so that no undue ratio of compression and consequent temperature can occur in any one stage. Increases in the air temperature which occur with compression must be reduced by intercoolers between each stage and by an aftercooler to reduce the final temperature to atmospheric temperature. Air-compressor valves are a fruitful source of trouble and must be removed at frequent intervals for inspection and cleaning. Air-receivers must be provided for storage and pressure equalization. Moisture that collects in the air-receivers must be drained at regular intervals. Lubrication of the air-compressor is particularly difficult, due not only to the high temperatures encountered, but to the danger of charging the air-receivers with an explosive oil-vapor. Disaster from this source is all too frequent.

Further disadvantages of the Diesel engine are that the power required to drive the air-compressor, usually taken as from 10 to 16 per cent of the engine rating, causes a lower mechanical efficiency and therefore a lower thermal efficiency since there is a less economical utilization of the fuel supplied. In addition, the expansion of the injection air in the combustion-chamber with the consequent absorption of heat, causes a refrigerating effect. To counteract this heat loss, a higher initial compression-pressure is necessary for obtaining the requisite temperature for ignition. The higher compression-pressure implies a lower mechanical efficiency, a heavier and more expensive construction, and an added difficulty in maintaining the higher compression-pressure under long service. The satisfactory operation of a Diesel engine requires complete combustion of the fuel. It is therefore essential that three quantities which influence combustion be varied in conformity with the variable conditions of speed and load imposed on the engine. The quantities are the quantity of fuel per charge, the quantity of injection air per charge and the fuel-valve lift and period of opening per charge. The usual practice is to vary the quantity of fuel per charge only. The quantity of injection air per charge is controlled only in a general way by the variation of the injection-air pressure through hand manipulation of an air-compressor regulating-valve, usually when the changed conditions are to remain for some time. Extended operation under no-load or light-load conditions is possible only with a diminution in the pressure of the injection air from that used at full load. The gear necessary for varying all three is necessarily very complicated and has been installed only on German submarine engines, where operators capable of maintaining the equipment are in constant attendance. In an engine designed for variable load and speed conditions such as are encountered in automotive service, it is essential that the quantities enumerated be varied by a suitable gear to conform with load and speed conditions. The

complication of such a gear renders the design impracticable.

Gas-Pressure Injection

In the gas-pressure method of injection the fuel flows or is injected into an auxiliary chamber which is in communication with the combustion-chamber. During the compression of the cylinder air a portion of the fuel within the auxiliary chamber is vaporized and burns, the resulting gas pressure within the auxiliary chamber forcing the fuel through small orifices into the combustion-chamber.

An example of gas-pressure injection is found in the Brons or Hvid high-compression oil-engine, illustrated in Fig. 7. The Brons engine is the original Dutch development. It is called the Hvid engine in this country from the name of the American patentee. The engine operates on the four-stroke cycle. On the suction stroke pure air is drawn into the combustion-chamber through the air intake valve *a*. The fuel supply pipe *b* supplies fuel to the chamber about the metering-pin *c* by gravity feed. During the greater part of the suction stroke the fuel-valve *d*, held open mechanically in conjunction with the air-intake valve *a*, permits the fuel to flow through the fuel-admission hole *e* into the fuel-injection cup *f* along with a small amount of air, which enters through the small opening *g* past the fluted stem of the fuel-valve *d*. The quantity of fuel admitted is regulated by the metering-pin *c*, which is in turn controlled by a governor. The air-valve *a* is closed at the end of the suction stroke. The fuel-valve *d* also is closed, covering the fuel-admission hole *e*. During the compression stroke all valves are closed and the air is compressed to 500 or 550 lb. per sq. in., thereby increasing its temperature to 1000 deg. fahr. Compressed air enters the injection cup *f*, through the small injection orifices at *h*, until the pressures are equalized. The pressure-temperature conditions within the cup are now most favorable for distilling off the lighter and more volatile components of the oil fuel. The fuel vapors ignite, due to the temperature existing, and the resulting high pressure within the cup forces the rest of the fuel out in a finely divided state, through the small orifices at *h*, into the air of the combustion-chamber. The pulverized fuel is vaporized and ignited by the temperature of the compression, and combustion takes place. The combustion releases heat and the expansion of the gases forces the piston out on its working stroke. As in any four-stroke cycle during the exhaust stroke, the exhaust-valve opens and the products of combustion are driven out by the piston. The Brons engine can be operated on the two-stroke cycle by mechanically injecting the fuel into the injection cup.

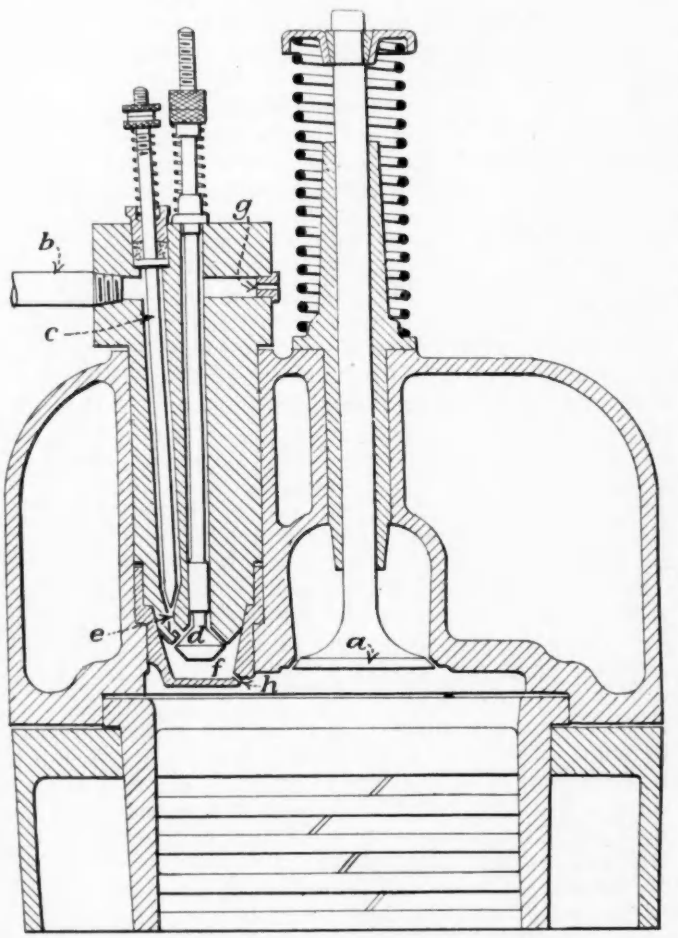

Fig. 7—Gas-Pressure-Injection System of the Hvid High-Compression Oil Engine

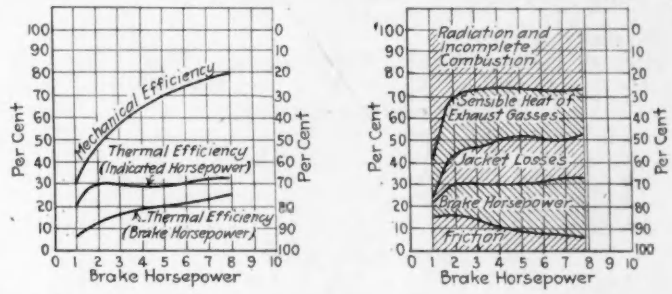

Fig. 8—Heat-Balance and Efficiency Curves for an 8-Hp. Single-Cylinder Farm Type Engine Operating on the Hvid Principle

Sears Roebuck & Co. have successfully marketed a series of horizontal, single-cylinder, farm-type engines which operate on the Brons principle. In Fig. 8 heat balance and efficiency curves are shown of one of their 8-hp. engines. Fig. 9 gives a comparison of the fuel consumption of this engine with that of a similar low-com-

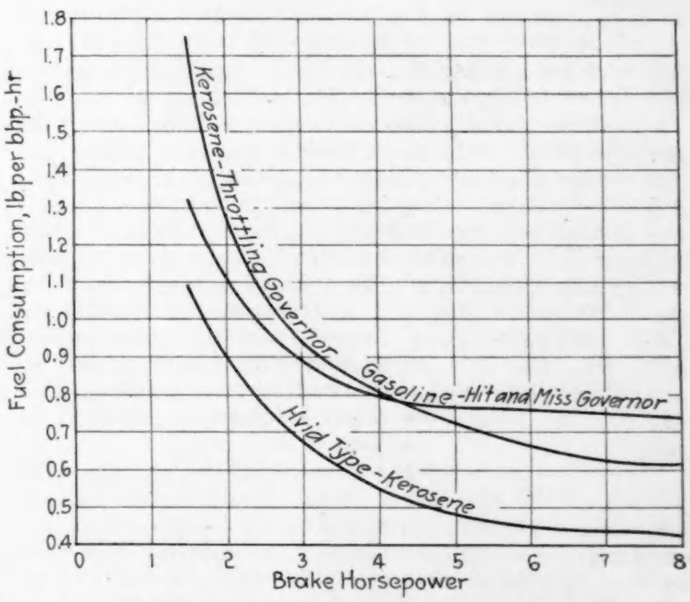

Fig. 9—Curves Showing the Comparative Fuel Consumption of an 8-Hp. High-Compression Engine and a Similar Low-Compression Engine

TABLE 1—PRICES OF SEARS ROEBUCK & CO. ENGINES DURING 1921

Brake Horse-power	Selling Price	Weight, lb.	Bore, in.	Stroke, in.	Cost per b.h.p.	Cost per lb.
Brons-Hvid Engines						
1½	$105.00	323	3	4½	$70.00	$0.325
3	155.00	615	3⅞	5½	51.66	0.252
6	260.00	1,100	5	7½	43.33	0.236
8	309.00	1,500	5¾	9	38.63	0.206
Average					50.90	0.259
Gasoline Engines						
1½	57.75	246	3¼	5	38.50	0.234
2½	93.50	483	4	6	37.40	0.193
5	136.00	777	5	7½	27.20	0.175
7	191.85	1,107	5¾	9	27.40	0.173
9	291.00	1,823	6½	11	32.33	0.159
12	365.00	2,433	7½	12	30.41	0.150
Average					27.64	0.180
Percentage of Greater Cost of Brons-Hvid over Gasoline Engines						
Prices for 1921					84.1	43.8
Prices for 1920					22.5	14.6

pression engine. Table 1 gives a comparison of their selling prices for Brons engines and for gasoline engines. The Brons engines are sold on the basis of their greater fuel economy, about 45,000 hp. being sold per year.

The success achieved by the Brons engines of Sears Roebuck & Co. can be attributed to the almost exclusive use of kerosene as fuel; the low loads imposed on the engines in farm service and the fact that the engines are usually operated at considerably less than their full-load rating; the low load factor as the engines are run but a few hours at a time and only a few days per week; and the elimination of crankcase dilution of the lubricating oil by the use of horizontal single-cylinder open-crankcase engines lubricated by oil-cups and grease-cups.

The Pittsburgh Filter & Engineering Co., located at Oil City, Pa., developed a Brons-Hvid engine which has given satisfactory service for over a year in supplying electrical power to its machine shop. The engine is a four-cylinder, vertical marine type, of 8½-in. bore and 12-in. stroke, rated at 100 b.hp. at 400 r.p.m. Under favorable conditions, the fuel consumption is 0.45 lb. per b.hp-hr., the fuel used at present is a Pennsylvania fuel oil of 28 to 42 Baumé gravity. Western fuel oils and kerosene are also satisfactory fuels. Aside from cleaning the injection cups occasionally, the engine requires no frequent servicing. There is no appreciable carbon deposit on the combustion-chamber parts. The piston does not appear to attain as high a temperature as in a Diesel engine. Crankcase dilution from fuel that escapes past the pistons requires that the lubricating oil be renewed occasionally. The exhaust is clear in color, showing only a very slight blue haze. More than 1½ years of experimental work was required on the proportions of the injection cup before the present satisfactory operation was attained. The data acquired by this laborious process are of very little use in constructing an engine of different cylinder volume or other considerable variance in conditions.

Claims are made for the Brons-Hvid engines that they possess all the advantages of the Diesel engine without the Diesel engine's principal disadvantage, the air-compressor. It is unquestionably the simplest high-compression engine to date; it will start and run on any free-flowing oil; its fuel consumption compares favorably with that of the Diesel; and it has remarkable "bull-dog" torque characteristics. Speeds of 1800 r.p.m. have been attained in experimental engines. Although the Brons-Hvid engine has been in existence nearly as long as the Diesel engine, it has achieved no success in Europe or America, except as distributed by Sears Roebuck & Co., as already stated. One of the objections urged against it is that the gas-pressure injection system, being entirely self-contained and automatic in operation, requires an exact relation in the proportions of the injection cup. A cup developed by long tedious cut-and-try methods until satisfactory for particular fuel, load, speed and engine conditions, will not operate satisfactorily with any considerable change of these conditions; nor are the data secured applicable to designing for different conditions.

The ideal conditions for the complete pulverization of the fuel cannot be attained in the Brons engine. This implies incomplete combustion with its attendant difficulties such as high explosion pressures; fouling of the combustion-chamber with the formation of carbon; and after-burning or delayed combustion with pitting of the exhaust-valves. A small quantity of fuel can be drawn from the injection cup into the combustion-chamber during the suction stroke. Some of this fuel will drain past the piston into the crankcase, causing a serious contamination and dilution of the lubricating oil with attendant ill effects on engine operation. Soft gummy carbon deposits are formed on the piston skirt, necessitating the removal of the piston for cleaning at regular intervals. Fuel remaining on top of the piston, instead of draining into the crankcase, will, during the compression, be "cracked" into hydrogen and free carbon. The hydrogen and other volatile constituents will ignite prematurely, causing excessive explosion-pressures which may range as high as 900 lb. per sq. in. An engine attaining such high explosion pressures must be made of heavier construction than one operating on the slow-burning or constant-pressure cycle.

The carbon accumulation may become so serious that the engine has to be dismantled to permit its removal. In some engines of this type there is no appreciable carbon accumulation. Poor speed and load regulation is an operating disadvantage of the Brons engine for many classes of service. The cutting off of the fuel supply by the metering-pin is not effective until after several charges have been discharged and burned. In starting or under sudden overload, partial charges may accumulate in the injection cup and, when discharged, will produce an excessive pressure that may result in completely wrecking the engine.

The Steinbecker engine,[3] which is a German gas-pressure-injection engine, has an auxiliary chamber, similar to an inverted bottle, which is in communication with the combustion-chamber. During the compression of pure air to 425 lb. per sq. in., a pump introduces some fuel into the neck of the chamber. The fuel is conveyed into the auxiliary chamber by the compressed air, which attempts to bring about an equalization of the pressure in the combustion and auxiliary chambers. The pressure-temperature conditions within the auxiliary chamber are such as to cause ignition of the fuel's volatile constituents. The gas pressure, resulting from this combustion in the auxiliary chamber, causes a reversal in the direction of the gases back into the combustion-chamber proper. The fuel, as it is introduced into the neck of the auxiliary chamber, is driven into the combustion-

[3] For a complete description of this engine see *Automotive Industries*, March 3, 1921, p. 501.

chamber in a finely divided state and is available for the combustion. The advantages claimed for the Steinbecker engine are similar to those claimed for the Brons, but the engine is considerably more complicated. Its disadvantages are, first, that an additional fuel is necessary for starting. In starting the engine, a very volatile fuel is injected by a special pump into the auxiliary-chamber throat; its combustion soon brings the chamber to a temperature that will cause ignition of the heavier fuel, whereupon the additional pump is cut out of operation. The main-pump plunger is immediately withdrawn upon completion of its stroke, thereby withdrawing the fuel in the discharge line and preventing an "after-drip" of fuel from the injection device. Second, there is difficulty in temperature control of the ignition chamber. In common with the Brons engine, the proportioning and temperature control of the ignition chamber for satisfactory operation under particular conditions as to fuel, load, speed and engine characteristics, can be arrived at only by long cut-and-try methods. Any considerable change in the conditions renders satisfactory operation impossible. Inasmuch as the operation of the Steinbecker automatic-injection process involves so many variables that are difficult to control, the engine must show satisfactory and consistent performance in actual service before it can be accepted as a successful design. The engine has undergone 7 years of experimental work and is now presumably ready for manufacture.

Mechanical Injection

When mechanical injection is utilized, the fuel is injected into the combustion-chamber by virtue of its own pressure produced by a mechanical pump. This type of engine offers a particularly attractive field for development in that it can utilize the same low-grade fuels as the Diesel engine, it can achieve as good if not better fuel economy and it can do this with a much simpler mechanism.

British firms have taken the initiative in developing this type of engine. Vickers, Ltd., has supplied most of the British submarines with mechanical-injection engines, and this company has gone extensively into the manufacture of engines of this class for merchant ships. It is stated that this type of marine engine totals over 500,000 hp. The recent motor-tanker Narragansett of the Anglo-American Oil Co. was fitted with two Vickers engines of 1250 b.hp. each. The engines are of the six-

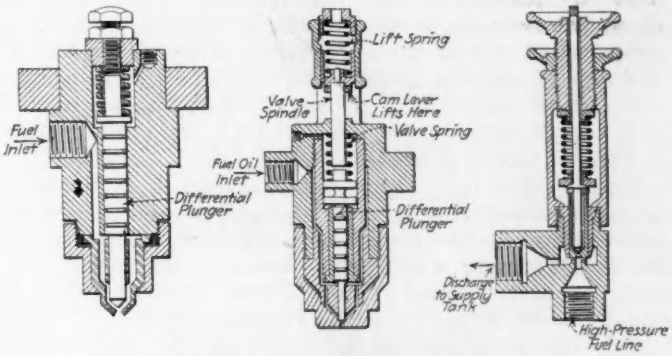

Fig. 11—Three Types of Valves

The Injection Valve at the Left Is Controlled by the Fuel-Pressure Difference and That in the Middle Has the Fuel Opening Controlled Mechanically by a Cam and a Lever. The Valve at the Right Is a Pressure-Regulating Valve.

cylinder, four-stroke-cycle crosshead type, with 24⅛-in. bore and 39-in. stroke, having a normal speed of 118 r.p.m. Their operation is that the fuel oil, which has been thoroughly cleaned of all grit by strainers, is delivered by a four-throw pump into a main pipe which supplies all the injection valves. The pump is arranged to maintain the fuel at a pressure of from 4000 to 4500 lb. per sq. in. The fuel-injection valve is mounted in the cylinder-head and consists of a spring-loaded miter-valve and stem. The stem is mechanically lifted by a cam and lever to permit the fuel to discharge into a small spherical chamber. The spherical chamber is provided with five holes 0.019 in. in diameter. The fuel, due to the pressure difference existing between it and the air of the combustion-chamber, flows into the spherical chamber when the valve stem is lifted, and is discharged through the small holes into the combustion-chamber as a fine spray. The temperature resulting from the 430-lb. per sq. in. compression, about 1000 deg. fahr., is sufficient to cause ignition and consequently combustion of the injected fuel. The time of injection and its period or duration can be varied in conformity with the engine load. The fuel consumption claimed for the Vickers engine is 0.381 lb. per b.hp-hr., but the engines of the Narragansett averaged 0.42 lb. per b.hp-hr. under the service conditions of a trial run. This figure is equal to that of the best Diesel engines.

Ruston & Hornsby, Ltd. builds a line of horizontal four-stroke-cycle engines suitable for stationary powerplant use. The line comprises 11 single-cylinder engines ranging from 15 to 170 b.hp. and twin-cylinder engines of from 100 to 340 b.hp. For burning the lowest grades of tar oil, it is necessary to inject a small quantity of a more inflammable fuel just prior to the main charge to insure ignition. This pilot charge, as it is termed, amounts to but 5 per cent of the total fuel used at full load. The injection-valve is not mechanically operated, as in the Vickers engine, but is actuated hydraulically by the pressure imposed on the fuel by the fuel-pump. The injection-valve for both pilot and main charges consists essentially of two spring-loaded differential pistons which uncover their valve-seats when the pressure of the fuel exceeds the resistance of the springs. The timing and duration of the valve-opening, the pressure and rate at which the fuel is injected and the quantity injected, are dependent upon the action of the fuel-pump and the governor. The fuel-pump action is influenced by the governor and by the timing and rate of motion imparted to the pump plungers by their driving cams. The governor controls a bypass valve which regu-

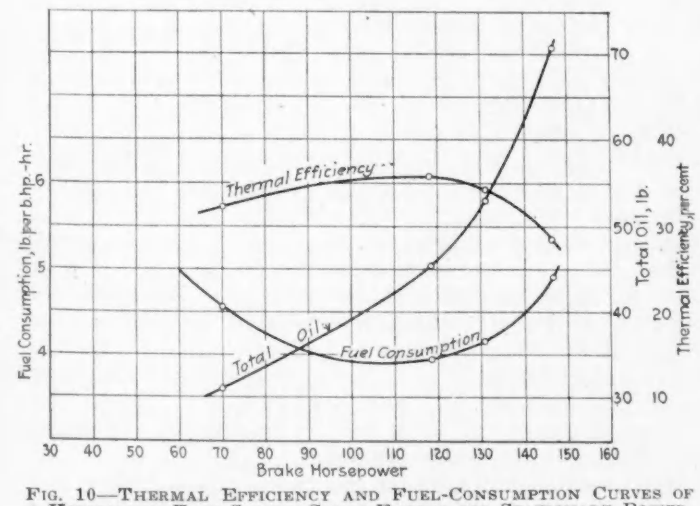

Fig. 10—Thermal Efficiency and Fuel-Consumption Curves of a Horizontal Four-Stroke-Cycle Engine for Stationary Powerplant Use

lates the quantity of fuel pumped per stroke. Fig. 10 shows the remarkable thermal efficiency and low fuel consumption of one of these engines. A characteristic of this type is that the thermal efficiency and fuel consumption remain fairly constant over a considerable range of load. An injection-valve, operating by fuel-pressure difference, is shown at the left of Fig. 11.

Crossley Brothers, Ltd., builds a series of horizontal single and double-cylinder stationary engines similar to those made by Ruston & Hornsby, Ltd. The engines will burn kerosene, fuel oils and tar oils. Their fuel consumption and thermal efficiency are approximately the same as those of the Ruston & Hornsby engines.

Only two prominent American firms have been building mechanical-injection engines for any length of time. Both engines are based on the same design and have given satisfactory operation in service. The De La Vergne Machine Co., which has been successfully constructing Diesel and hot-bulb engines for the last 26 years, has recently developed mechanical-injection engines of both the vertical marine and horizontal stationary types. The engine operates on the four-stroke cycle. Pure air is drawn into the cylinder on the suction stroke; the air is compressed to a pressure of 330 lb. per sq. in. on the compression stroke. At about 17 deg. of crank travel before the end of the compression stroke, the fuel is injected into the combustion-chamber through two injection-valves that are fitted on opposite sides of the combustion-chamber. The combustion-chamber is in the form of two half-cones with their bases fitted together. The injection-valves are located at the apex of each cone and therefore are opposite each other and discharge toward the common center, which is at the bases of the cones. The fuel, at a pressure of from 1500 to 2000 lb. per sq. in. produced by a cam-operated pump, causes the two spring-loaded injection-valves to open. The injected fuel, being in a finely divided state, is ignited by the heat of compression. Due to the early fuel injection, the initial combustion occurs at constant volume, the pressure rising from that of the compression to 500 to 550 lb. per sq. in. Additional combustion then occurs at constant pressure, as in the Diesel engine. The expansion and the exhaust strokes are completed in the usual manner. The use of the two injection-valves and the peculiar form of combustion-chamber, which is a distinctive feature of oil engines designed by the late William T. Price, is considered advantageous in the atomization of the fuel and in the creation of additional turbulence in the combustion-chamber. The combination of the Otto and the Diesel cycles with combustion at both constant volume and constant pressure is a characteristic of most mechanical-injection high-compression engines and has been termed the "dual combustion cycle." A pressure-volume card is shown in Fig. 12. The Blackstone engine mentioned under low-compression engines also operates on the dual combustion cycle.

Table 2 gives fuel-consumption data obtained in a test of a De La Vergne Type S-I horizontal two-cylinder four-stroke mechanical-injection engine of 17-in. bore and 27.5-in. stroke, rated at 200 hp. at 200 r.p.m. The compression-pressure of cylinder No. 1 was 322 lb. per sq. in. and that of cylinder No. 2 was 318 lb. per sq. in. The maximum pressure at full load was 540 lb. per sq. in. for cylinder No. 1 and 545 for cylinder No. 2. The test was made on Oct. 5, 1920. Fuel oil of 24-deg. Baumé gravity was used. It is stated that the engine will carry 25-per cent overload with no trace of smoke or color in the exhaust; at 30-per cent overload the exhaust begins to become smoky. The fuel consumption obtained compares favorably with that of a Diesel engine.

TABLE 2—DE LA VERGNE TYPE S-I ENGINE TEST

Rated Load, per cent	Brake Horsepower	Fuel Consumption, lb. per b.hp.	Engine Speed, r.p.m.
100	200	0.420	200
110	220	0.421	200
75	151	0.410	202
50	101	0.438	204

Although this firm had an established line of Diesel engines, these were discontinued and the building of mechanical-injection engines was undertaken. This was because the latter have, in addition to the advantages of the Diesel engine, such as low fuel-consumption with a wide range of fuels and loads, immediate starting and no standby losses, an inherent advantage of simplicity resulting from the elimination of the air-compressor and air-injection apparatus.

The Ingersoll-Rand Co. is the sales agent for the Price-Rathbun engines which, having the Price injection and combustion system, possess many features in common with the De La Vergne engines, and naturally the same advantages are claimed. The type developed is a vertical four-stroke cycle, which is adapted to either marine or stationary service. The fuel-consumption guarantee is the same as that of the De La Vergne Co., namely, 0.45 lb. per b.hp-hr.

As compared with other high-compression engines, mechanical-injection engines hold possibilities for the attainment of the ideal conditions for optimum pulverization of the fuel without the complication or disadvantages of air-injection or gas-pressure injection. They also accomplish the elimination of the air-injection apparatus of the Diesel engine which includes (*a*) The air-compressor with its pistons, connecting-rods, cranks, numerous valves and controls, intercoolers, aftercooler and oil separators; (*b*) the high-pressure air-pipes, injection air-receivers, bleeder-valves for discharging moisture and their connections; (*c*) the air type of injection-valve with its high-pressure stuffing-box on the valve spindle and (*d*) the various regulating devices for variable-load conditions.

The benefits secured are simplicity of construction with lower capital and operating costs, continuous reliability with lower operating costs and higher mechanical efficiency, because 10 to 16 per cent of the Diesel engine's power is required to drive the air-compressor. This saving in power results in lower fuel-consumption, particularly at fractional loads. Further advantages are greater ease in operation and maintenance because the operator has to observe only the fuel and lubricating systems and has no high-pressure system to watch; and increased safety in that there are no high-pressure air-bottles to explode. Also, the injection-valves cannot stick open, due to the pressure stuffing-box, and so allow the injection air to discharge the fuel prematurely, which would result in dangerous preignition. When the air-injection valve sticks open for some time, the injection air is soon lost and causes stoppage of the engine. Unless there are reserve air-receivers available, which are charged to full-injection pressure, the engine cannot be run until the receivers are charged. With large Diesel units auxiliary power is provided for pumping up the air-receivers; otherwise the engine would have to be motored for building up the injection-air pressure.

Further benefits are that the mechanical-injection engine has a simpler mechanism for governing under vari-

able loads since, in the Diesel engine, the pressure and quantity of injection air must be varied as well as the quantity of fuel per charge. This is mentioned in the discussion of the disadvantages of the Diesel engine. The time, duration and rate of injection, and therefore the quantity of fuel injected, can be controlled easily in the mechanical-injection system, through simple governing devices. There is also a saving in the amount of space required by the engine.

In the Diesel engine the expansion of the high-pressure injection air in the combustion-chamber produces a refrigerating effect which chills both the combustion air and the fuel. This refrigerating effect necessitates the use of a higher compression to secure a high enough temperature for ignition. Mechanical injection, having no injection air, can utilize a lower ignition-temperature and therefore a lower compression-pressure. The benefits secured are (*a*) lighter and less expensive construction; (*b*) a higher mechanical efficiency; (*c*) easier starting and (*d*) less difficulty in maintaining the compression under long continued service. The mechanical-injection engine gives better speed-regulation for variable loads, particularly in comparison with gas-pressure-injection engines like the Brons and Steinbecker, and it will operate at lower speeds and loads than the Diesel engine. Further, it has a lower fuel-consumption and an increased thermal efficiency. In the past few years of development of the mechanical-injection engine its fuel-consumption has been consistently lowered so that it is now equal to, and in some instances lower than, that of the Diesel engine. Its fuel-consumption is consistently lower than that of the gas-pressure-injection engine.

As compared with other high-compression engines, the disadvantages of the mechanical-injection engine have been due primarily to their inability to attain the ideal conditions for optimum pulverization. These include the objections that

(1) The fuel-injection is initiated as a dribble instead of as a pulverized spray
(2) Pulverization is not carried far enough toward achieving atomization and the fuel is discharged in the form of globules instead of as particles without finite mass
(3) The pulverization is not uniform throughout the period of discharge
(4) The injection ceases with a dribble and after-drip
(5) The globules of fuel resulting from the dribbles and after-drip are deposited on the walls of the combustion-chamber, "cracking" takes place with the formation of carbon deposits and the combustion-chamber must then be cleaned of carbon
(6) Vaporization and gasification are delayed by poor pulverization. In usual designs the fuel injection is begun 18 deg. ahead of compression dead-center. The globules of fuel are subjected to the heat of compression and to the heat resulting from early combustion for a sufficient time, presumably, to produce vaporization
(7) Explosive constant-volume combustion is secured as a result of the early injection of the fuel, instead of the constant-pressure combustion of the Diesel engine. The compression-pressure of the mechanical-injection engine may be considerably lower than that of the Diesel, but its explosion pressure may exceed that of the Diesel; therefore, the mechanical-injection engine must, in many designs, be constructed for higher pressures than the Diesel. This implies a heavier engine
(8) After-burning, or delayed combustion, occurs with poorly pulverized fuel because there is insufficient time for the globules to become vaporized and ignited until late in the power stroke
(9) The intermixture of the gasified fuel and oxygen cannot take place as rapidly and effectively as in the Diesel engine, where the injection air produces considerable turbulence
(10) The complication of the pump that is necessary. With automatic injection valves operated by fuel-pressure difference, the pump must produce instantaneous high pressure and, in conjunction with a governor, control the timing and duration of the injection, as well as meter the small quantity of fuel to be injected
(11) The quantity of fuel injected is so small that the slightest leakage in the fuel system will cause the loss of the high pressure and prevent the operation of the injection-valve
(12) The quantity of fuel between the fuel-pump and the injection-valve must be at a minimum in order that the elasticity of the fuel and of its container cannot influence the manner of injection

Summary

The thermal efficiency of the internal-combustion engine is admittedly superior to that of any other type of prime-mover. A survey of the existing types of internal-combustion engine shows that high-compression engines have the greatest thermal efficiency and are capable of operating on low-gravity oils as well as the more volatile fuels. Developing an engine to meet the small-power high-speed requirements of automotive engines necessitates that the engine be of extreme simplicity to insure ease of operation and continuous reliability. In the Diesel engine the complications of the air-compressor and air-injection system are such as to render highly impracticable the engine's development for that class of service. The great amount of development work expended on the Brons engine since its inception, with the meager success achieved, indicates that the engine has many inherent disadvantages that do not recommend it as being possible of development for automotive service. The thermal-efficiency of the mechanical-injection high-compression engine equals that of the Diesel engine and is superior to that of any other type of prime-mover. In the development of the commercial mechanical-injection engine the path of progress can be traced from the efforts to eliminate the disadvantages of the hot-bulb engine, rather than attempts to substitute mechanical for air injection in the Diesel engine. Naturally, the mechanical-

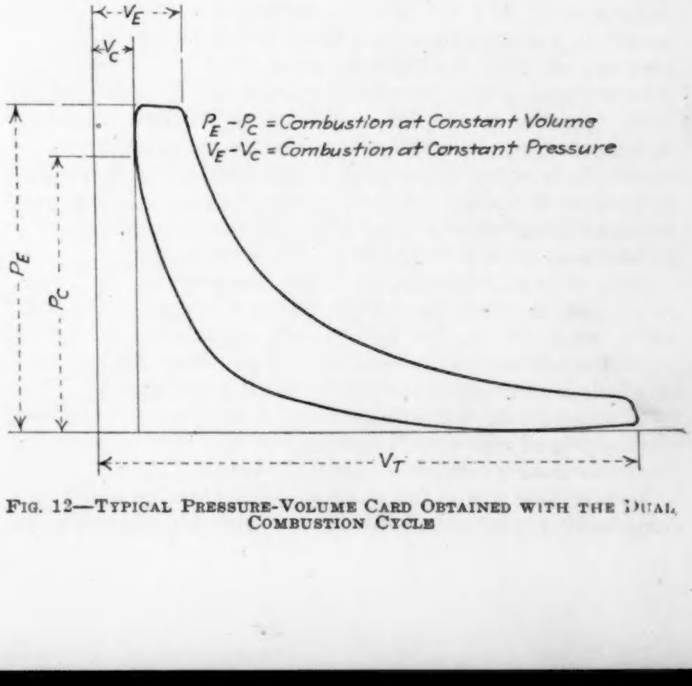

Fig. 12—Typical Pressure-Volume Card Obtained with the Dual Combustion Cycle

injection engine inherits many of the defects of the hot-bulb engine, but marked progress is being made in the elimination of these difficulties.

An analysis of the disadvantages of mechanical-injection engines will show that there are no inherent disadvantages that cannot be eliminated by intensive and intelligent development. The undesirable characteristics retained in the present designs can be attributed to the inability of their constructors to achieve the conditions for optimum pulverization and to a lack of fundamental knowledge concerning the handling of high-pressure fluids.

The final criterion of the economic value of any device for human consumption is its comparative overall cost under continued service, and its ability to meet the particular requirements and desires of its users. In the comparisons given of the various engines, the generalizations made do not indicate the relative value of the advantages and disadvantages for specific classes of work. The problem is complicated further by the kinds of service for which automotive engines are used; they may provide motive power for transportation on land, on sea and in the air and for an infinite variety of isolated powerplants.

The mechanical-injection engine is inherently simpler than the present type of automotive engine; its higher pressures simply require a stronger, more sturdy construction. The success achieved by mechanical-injection two-stroke-cycle engines shows the possibilities of developing two-stroke-cycle engines for automotive purposes with the mechanical-injection system; whereas, the two-stroke-cycle low-compression engine has been unsuccessful except in a very few instances. Estimates of the comparative manufacturing costs of mechanical-injection and present automotive engines have not been attempted, due to the many variables affecting the respective costs. For instance, the quantity to be produced is a very decisive factor in manufacturing costs. That high-compression engines can be sold at a higher price than similar low-compression engines on the basis of their lower fuel costs, is evidenced by the success of Sears Roebuck & Co. in marketing their Brons engines.

The engine-fuel situation is such a new development that builders have but recently realized the scope and importance of the problem. There was no warning until the condition was established; therefore practically no development work has been attempted in this country to produce a satisfactory high-compression engine for automotive use. Until recently, the demand for automotive products was so great that the manufacturers were concerned only with problems of production; they would not contemplate scrapping existing designs. Men with sufficient training to inspire confidence as to their ability to develop high-compression automotive engines have secured their experience with firms producing large stationary and marine engines. These organizations have retained their men so that they have not been available to the automotive industry. Mechanical-injection high-compression engines operating on low-grade fuels can be developed to meet the requirements of automotive purposes, particularly for heavy-duty work such as truck, tractor, railroad, motor-car and motor-boat service and the infinite variety of small isolated powerplants. The data acquired in the development of engines for the services outlined above will render further development for motor-car use a matter of detail progress.

As a motive power for airplanes and airships, the high-compression mechanical-injection engine is believed to be a necessity for the development of commercial aviation. To quote the National Advisory Committee for Aeronautics:

> It is asserted that the capital investment, maintenance charges and fuel costs are all very high in the case of the present aircraft engines and must be lowered materially before the cost of power can be reduced to figures which will make possible the extensive development of commercial and pleasure aviation. The shortage and high cost of aviation gasoline, as well as the complication and relative unreliability of the carburetion and ignition systems, emphatically indicate the necessity for the development of an engine which will operate by direct hydraulic injection of low-grade fuel, with a compression sufficiently high to insure automatic ignition. The committee feels that the early development of an engine of this type is one of the most important technical problems involved in the growth of commercial aviation in this country.

The increased safety resulting from the use of a less volatile fuel than gasoline and the lower fuel-consumption of the mechanical-injection engine are very desirable features. Diminishing the weight of fuel required permits an increase in the pay-load.

It is believed that the preceding comparisons of the various types of internal-combustion engine indicate the advisability of developing a high-compression mechanical-injection engine for adaptation to automotive uses. It was with this conviction that the development work which follows was undertaken.

DEVELOPMENT WORK

The experimental work in the development of a mechanical-injection system for a high-compression engine has been done in the mechanical laboratory of the Carnegie Institute of Technology, Pittsburgh, through the kindness of William E. Mott, director of the division of science and engineering, and C. L. Willibald Trinks, professor of mechanical engineering.

A small surface-ignition engine built by the National Transit Pump & Machine Co., Oil City, Pa., was used, after increasing the compression-pressure to 500 lb. per sq. in. by the substitution of a new piston and a new cylinder-head. The engine, which is illustrated in Fig. 13, is a vertical two-stroke-cycle two-port type, with crankcase precompression. The bore is 5½ in. and the stroke is 6 in., although the effective stroke is 4¾ in. after making allowance for the ports. The compression of 500 lb. per sq. in. was selected for the initial tests, as it can be decreased when desired by simply raising the cylinder-head. The compression necessary for auto-ignition depends upon the ignition temperature of the fuel; the temperature from the compression must be high enough to insure starting with the engine cold. The lowest compression that will permit cold starting is, from present practice, about 350 lb. per sq. in. Fig. 4 gives temperatures resulting from different compression-pressures. The compression ratio necessary for producing a particular compression-pressure can be computed from the equation $P \cdot V^n = $ a constant; where n equals 1.25 to 1.31, depending upon engine conditions.

The determination of the ignition temperatures of fuels by ordinary laboratory equipment is unsatisfactory in that such variables as the manner and quality of injection and the temperatures of the combustion-chamber walls are not considered. Apparatus consisting of a combustion-chamber fitted within a gas furnace and equipped with an injection-valve, and pressure, temperature and

time recording devices, has been used by the British Admiralty Experimental Laboratory[4] for determining the ignition temperatures and the time-lag which elapses from the moment of injection until a pressure rise is indicated in the combustion-chamber.

Mechanical-injection can be accomplished by variable pressure, in which the fuel-injection valve-opening is controlled by fuel-pressure difference; or by constant pressure, in which the fuel-injection valve-opening is controlled mechanically by a cam and lever and the fuel pressure is constant.

The Variable-Pressure Method

The variable-pressure type of injection-valve, an example of which is shown at the left of Fig. 11, consists of a spring-loaded differential plunger that opens when the fuel pressure exceeds the spring resistance. The fuel-pressure variation producing injection is dependent upon the action of the fuel-pump and governor, which action should cause the injection to take place in accordance with the conditions that were stated for optimum pulverization. In addition, the quantity of fuel to be injected must be metered accurately and the time and duration of injection must be under control. The small quantity of fuel to be injected previous to each working stroke is responsible for some of the principal problems in the design of a small high-speed engine. In the experimental engine under discussion the quantity of fuel injected per revolution at full load is about 0.1 cc.

The usual method of metering the fuel is for the governor to hold open the suction-valve or a bypass-valve during the initial portion of the pump stroke and, upon the closing of the valve, the fuel is pumped into the in-

[4]This is described in a paper entitled Fuel Oil in Diesel Engines which was presented by Prof. C. J. Hawkes at a meeting of the North-East Coast Institution of Engineers. An abstract of the paper was published in *Engineering* (London), Dec. 3, 1920, p. 749, and Dec. 10, 1920, p. 786.

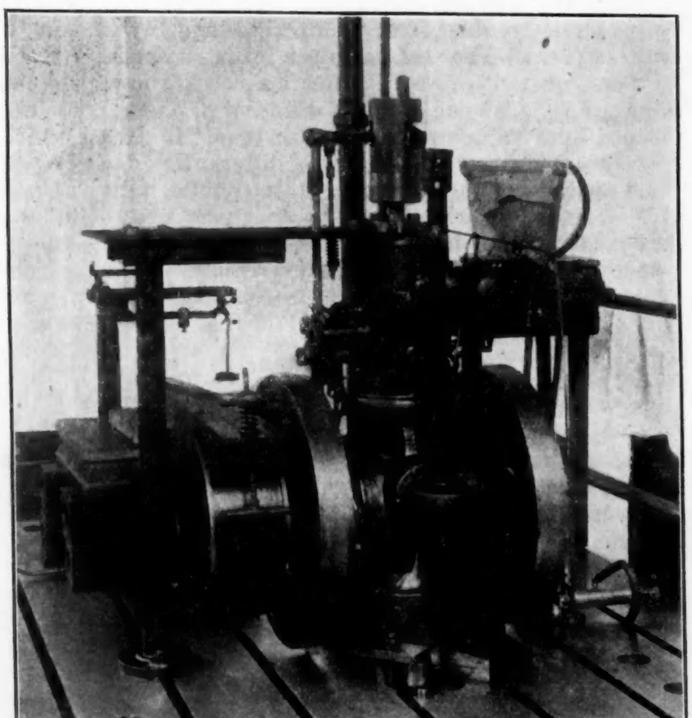

Fig. 13—The Vertical Two-Stroke-Cycle Two-Port Type Engine with Crankcase Precompression upon Which the Experimental Work Described in the Paper Was Done

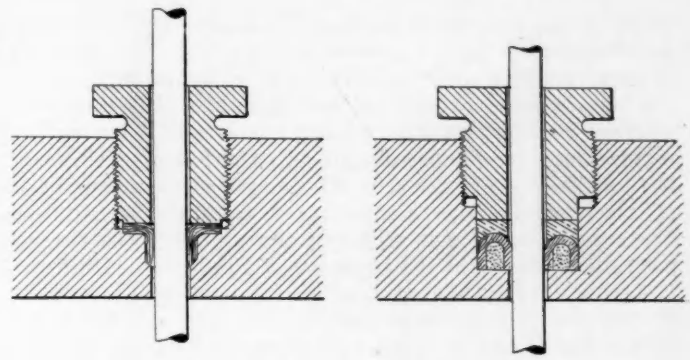

Fig. 14—Two Types of Leather Valve Packing
The Packing at the Left Is of the Flange Type While That at the Right Is What Is Termed a U-Packing.

jection-valve. Another method consists of varying the stroke of the pump plunger by a wedge actuated by the governor. With accurate metering by the fuel-pump the problem then consists in discharging exactly the metered quantity of fuel from the injection-valve, and in discharging uniform quantities of fuel for uniform speed and load conditions.

Conditions which may interfere with or prevent the injection of the metered quantity of fuel, are the (*a*) elasticity of the fuel; (*b*) elasticity of the fuel-containing system; (*c*) elasticity of entrained or entrapped air and (*d*) leakage within the fuel system. The elasticity of the fuel at the pressures used is approximately 2 per cent. This condition makes it necessary that the quantity of fuel in the system, between the pump and the injection-nozzle, be the least possible.

To overcome elasticity in the fuel-containing system it is obvious that the fuel pipes, connections and the like must be made heavy enough to insure absolute rigidity. The poor injection action of many surface-ignition engines can be attributed to this condition. Air gets into the fuel system by being entrained with the fuel, by leakage past the injection-valve during the compression stroke and by leakage during the suction stroke of the fuel-pump. It may be entrapped during the assembly of the fuel system, or it may accumulate while the engine is idle. Air is a very slippery material and will apparently work into the fuel system when the system is so tight that no fuel leakage can occur. Air is so highly elastic that a small bubble in the fuel system will become a minute volume during the working stroke of the fuel-pump but, when the pump starts on its suction stroke, the bubble expands and prevents the pump from drawing in a full fresh charge of fuel. During the following working stroke of the pump, either no fuel will be discharged or the quantity will be less than that desired. This produces an erratic "hit-or-miss" injection action that may cause the injection of an excessive quantity of fuel, thereby producing abnormal pressures in the cylinder. Provision must be made for draining the air from the highest point in the fuel system, preferably in the injection-valve. The fuel system must be arranged so that the air will not accumulate at any other point.

The quantity of fuel in the system, particularly in the injection-valve, should be a minimum to secure as complete and rapid a discharge of the fuel and entrained air as possible. Fuel leakage occurs around the pump-plunger packing-gland and around the packing-gland or similar device of the injection-valve stem. In the injection-valve shown at the left of Fig. 11 the valve-stem is lapped

to its guide and there is slight leakage. High-pressure packing-glands are a source of trouble. A form of plunger packing used satisfactorily in hydraulic machinery at pressures exceeding that encountered in injection engines is shown in Fig. 14 and consists of Vim leather molded into the shapes shown. In the flange packing at the left the nut clamps only the flanged portion; the body of the packing has clearance so that the liquid can surround and force it tight on the plunger. In the U-packing at the right the pressure of the liquid within the U forces the leather against the plunger and the walls of the receptacle, thus sealing the joint. The space within the U can be filled with hemp to prevent collapse of the packing when there is no pressure acting. Leakage may occur at the suction or discharge check-valves, which are usually placed in multiple. Poppet, cylindrical and ball-check valves are shown in Fig. 15.

A form of pipe connection used satisfactorily in Diesel engine practice is shown in Fig. 16. The male part is made of brass or copper, the tubing being brazed into it.

The simplicity of the injection-valve is the advantage of the variable-pressure method; the operation of the valve is entirely automatic, avoiding the complication of valve-gear. The quantity of fuel for each charge can be metered accurately by simple and reliable mechanism incorporated within the fuel-pump, but the disadvantage of the method lies in its inability to discharge the desired quantity of fuel at the correct time and in the manner necessary for achieving complete combustion. The reasons for this inability have been enumerated. Of these (*a*) the elasticity of the fuel and (*b*) the elasticity of the fuel-containing system can be partially obviated by proper design but (*c*) the elasticity of entrained and entrapped air and (*d*) leakage within the fuel system are more difficult to control. Due to the small quantity of fuel to be discharged the presence of any air will interfere with or prevent regular injection of the fuel, and a very slight leakage can easily equal the quantity of fuel it is desired to inject. These adverse conditions make the variable-pressure method practicable only for large slow-speed single-cylinder engines, where the amount of fuel to be injected is a relatively large quantity. The method is in use on all surface-ignition engines and is therefore to be found on many high-compression engines developed by companies building both types. The high-compression engines using this method are usually of the slow-speed single-cylinder type, which can be assembled also as twin-cylinder engines. The conclusion arrived at is that this method is not satisfactory for high-speed multiple-cylinder engines suitable for automotive purposes.

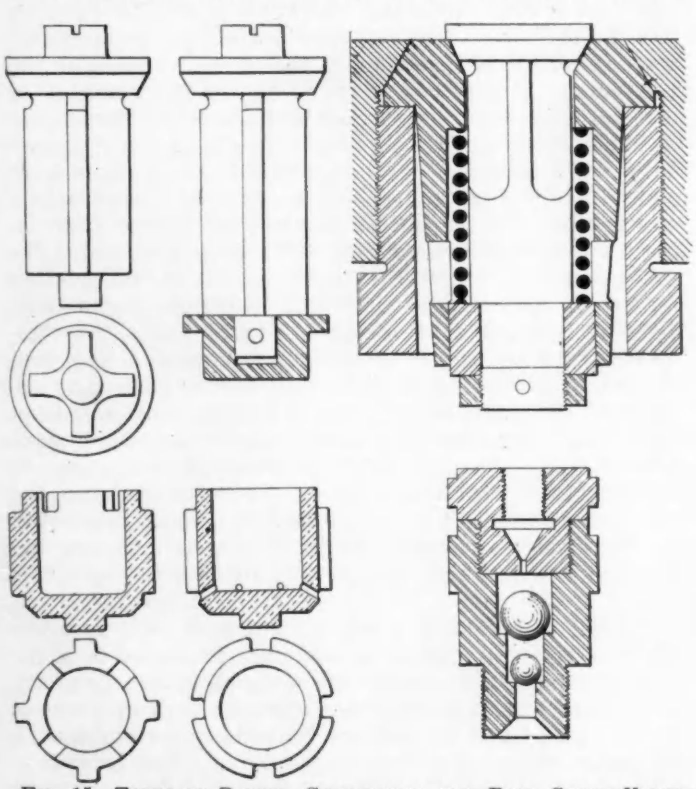

FIG. 15—TYPES OF POPPET, CYLINDRICAL AND BALL CHECK-VALVES USED IN INTERNAL-COMBUSTION ENGINES

THE CONSTANT-PRESSURE METHOD

The fuel is supplied to the injection-valves of all the cylinders from a single main supply line which is maintained at constant pressure. The fuel-injection valve shown in the center of Fig. 11 is mechanically operated by a cam and lever; the metering of the quantity of fuel for injection and the time and duration of the injection are under control of the governor acting on the fuel cam and lever. The fuel-pumps are designed to pump two or three times the quantity of fuel necessary for full-load conditions. The fuel-pumps are made preferably from a forged-steel block; the pump cylinders and valve-seats being machined in the block. Packing-glands for the pump-plungers have been shown in Fig. 14 and the check-valves in Fig. 15. Due to the high pressures that the pumps work against, the pump shaft and the eccentrics must have liberal bearing areas and positive-pressure lubrication. The use of several plungers of small cross-sectional area reduces the bearing loads. The pressure of the fuel is maintained constant by a pressure regulator holding suction or bypass-valves open during a portion of the pump stroke, or by a spring-loaded relief-valve similar to the pressure-regulating valve shown at the right of Fig. 11. The flat seat of the pressure-regulating valve gives less trouble from chattering and wear than a mitered seat. The seat parts are made of hardened steel.

The constant-pressure injection-valve shown in the center of Fig. 11 consists of a differential plunger, spring-loaded by a valve-spring and also by a lift spring which forces a valve spindle onto the differential plunger. A cam-operated lever lifts the valve spindle, relieving the lift spring and permitting the fuel pressure to lift the differential plunger against the valve-spring. The movement of the differential plunger removes the injection-valve from its seat and permits the fuel to be injected into the combustion-chamber. The fuel cam action controls (*a*) time of injection; (*b*) lift of the injection-valve; (*c*) duration of injection; (*d*) quantity of fuel injected and (*e*) manner of injection. Variations in the time of injection can be accomplished by angular advancement or retardation of the fuel cam. The lift of the injection-valve and the duration of the injection are determining factors in the quantity of fuel that will be injected. The injection-valve lift, determined by the fuel-cam lift, must be sufficient to permit free injection of the fuel without dribbling. The duration of injection depends upon the angular period of the fuel-cam. As the fuel pressure is constant, the injection-valve lift should remain practically constant during the period of injection. Varying the quantity of fuel for different load conditions should be accomplished by varying the duration of the injection period; in other words, varying the cut-off ratio, or the length of the constant-pressure line. The rapidity with which the injection-valve opens

and closes is a determining factor in attaining the conditions of maximum pulverization. With the injection-valve filled with fuel maintained at constant pressure and the fuel cam acting directly on the injection-valve, the injection action can be much more rapid than with the variable fuel-pressure generated by a cam and pump removed from the injection-valve.

The constant-pressure method is not subject to the adverse conditions which interfere with injection in the variable-pressure method. The constant fuel-pressure obviates trouble from the elasticity of the fuel and its containing system and also does not permit the entrapped air to expand and cause erratic injection. Any leakage that occurs in the fuel system with the variable-pressure method is a wastage from the metered quantity of fuel that it is desired to inject. In the constant-pressure method the metering is done within the injection-valve so that leakage in the system does not affect the quantity to be injected.

In view of the considerations stated, the constant-pressure method is preferable for small high-speed engines in that the exact metered quantity of fuel is positively injected at the desired time, and in that uniform quantities are injected for uniform speed and load conditions. The experimental work has shown consistent progress but has not advanced to a point where further disclosure of methods and results is advisable at this time.

The imperative necessity for the conservation of the world's petroleum resources is too well realized to require repetition. It behooves the automotive industry to have an open mind in considering new types of engine for development into more economical engines for the future. It is hoped that the paper submitted has been instrumental in securing recognition of the mechanical-injection high-compression oil engine as the logical selection, of the known types of internal-combustion engine, for development to meet the requirement of automotive equipment. The development problem is a big one, but it is certainly not one that the automotive industry, which has been the very embodiment of the spirit of progress, need fear to attack. America leads the world in the production of automotive equipment, but to lead in the development of more economical types will require the cooperation of the entire automotive industry, including its business men and its engineers.

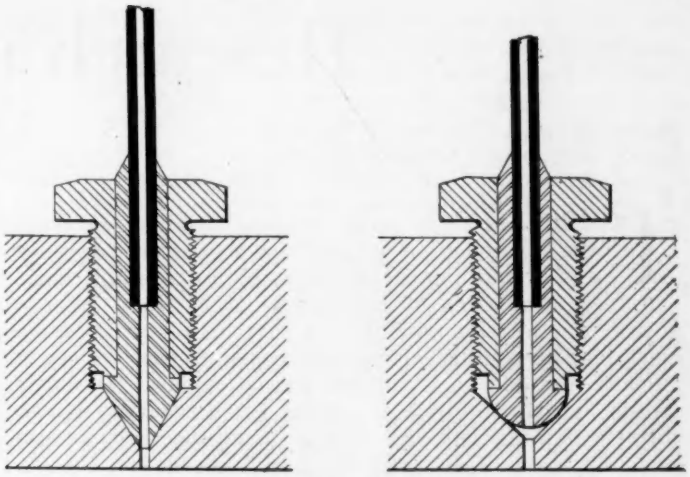

FIG. 16—A FORM OF PIPE CONNECTION USED WITH GREAT SATISFACTION IN DIESEL ENGINE PRACTICE

The encouragement and assistance rendered by the administrative and instructional staff of the Carnegie Institute of Technology is gratefully acknowledged. Acknowledgment is due also to Stanley L. Connell for his collaboration in the development work and the preparation of this paper.

THE STRAIGHT-EIGHT ENGINE

SO much has been suggested and reasoned concerning the fundamental principles which led to the recrudescence of the eight-cylinder-in-line engine for racing purposes that it is most interesting to learn the views of those engineers in France who sponsored the movement. The straight-eight appears to have been adopted because it is well balanced; allows of a higher number of revolutions than is possible with four cylinders of equal displacement; because the lighter reciprocating parts, such as valves, though multiplied in number, are less highly stressed, and hence should prove more reliable; and because with this design cooling, in the sense of the avoidance of distortion, is more thoroughly effected. It would seem, therefore, that some of the engineers on this side of the Channel were correct in their assumptions. Others, however, are very definite in their preference for the large cylinder on the score of the possibility of lighter construction. There seems to be the making of an illuminating controversy as to whether the car of the future should have a small high-speed high-efficiency engine with many cylinders, or the larger slow-speed engine of the four-cylinder variety. If the opinion of car users as a whole were taken, and the effect of the Treasury tax on bore were for the moment put aside, we think that opinion would be more or less evenly divided. One side would favor the large soft engine, characteristic of so many American cars, with its top-speed capabilities, while the other would embrace the high-speed engine and the four-speed gearbox, which is common to British medium-power car design.—*Autocar* (London).

BRITISH AND AMERICAN INCOME TAX

THE British Royal Commission on Income Tax estimates that there were 3,406,000 taxable incomes in Great Britain in 1918, producing income and supertax amounting to £272,816,000. These figures compare with 1918 Internal Revenue returns of 4,425,114 taxable incomes in United States, yielding a tax amounting to $1,127,721,835.

COOPERATION AND COORDINATION

COOPERATION means essentially the working together of equals, with none surrendering the right to independent action should self-interest demand it. Coordination means the surrendering of certain individual prerogatives and the submission to the authority of a superior for the common good.—*Electrical World.*

Research in Industry

By L. A. Hawkins[1] Semi-Annual Meeting Paper

THERE are three steps common to all researches; the statement of a question, the performance of one or more experiments to obtain relevant data, and the formulation of conclusions from those data to answer the question wholly or in part. And yet the term research covers multifarious activities different widely in purpose, methods and importance. Compare Millikan's classic experiment with oil drops for measuring by an entirely new method the charge on an electron, an experiment requiring imagination, a high degree of resourcefulness and delicacy of manipulation, with results of enormous interest to science but of no immediate commercial utility, with such an investigation, for instance, as one now being conducted at the University of Illinois on fatigue effects in 3-per cent nickel steel, an investigation for which the chief requirement is an adequate testing equipment, and the results of which will be of immediate practical utility to the designers of steam turbines. A definition of research that covers both of these investigations must obviously be general in its terms.

Research of a type most useful to one industry may be largely inapplicable to another. This paper will confine itself to research in electrical manufacture and will leave it to you automotive engineers to judge how far our experience is applicable to the automobile industry.

The General Electric Co. was among the pioneers in this country in the field of industrial research. The laboratory at Schenectady, starting from small beginnings about 18 years ago, has grown until we now have about 300 people on our payroll, about one-quarter of whom are trained pysicists, chemists, metallurgists and engineers. In the limited time at my disposal it would be futile to attempt to describe all the different kinds of work in progress in the laboratory. There would be time to hardly more than mention them, and the result would be about as interesting as a catalog. So I shall try to present the picture rather impressionistically, referring to specific lines of work only by way of illustration.

Classes of Research

Our work in the laboratory naturally divides itself into two general classes, distinguished from each other by the immediate objects in view, by the methods by which those objects are sought, and also, to some extent, by the type of mind best adapted to pursue those objects. In the first place, we have researches which have a concrete object in view, a definite commercial goal in sight. On the other hand, we have researches more in the nature of pure science, the object of which is to extend our knowledge of things electrical, with no concrete object in sight. I can illustrate those two classes of work by two of the best known achievements of the laboratory, both in the incandescent-lamp field; one the production of drawn tungsten, the other, the gas-filled lamp. Both of these, as you know, were of great importance, and yet the kind of work which led up to them was totally different in the two cases.

When tungsten lamps were made from the fragile, costly, squirted filament it was obvious to almost anyone that the real way to make a tungsten lamp was to get tungsten in the wire form and draw it; the difficulty was to find out how to do it. A great many metallurgists tried and failed, but Dr. Coolidge succeeded. Now he succeeded by persistent, resourceful experiment, trying this, that and the other thing; when it failed, repeating, changing one thing or another; making a little progress here, learning a little something from a failure there, but always moving forward toward the goal which was more or less clearly visualized from the start, until finally he had the thing he was after, drawn tungsten.

The origin of the gas-filled lamp was totally different. Dr. Langmuir had been conducting what you might call an academic investigation of the evaporation of hot tungsten, and he found an interesting scientific fact, that the rate of the evaporation of tungsten is affected to a remarkable extent by gas pressure. While he was on that investigation he was requested by the heating-device department to determine the laws governing the loss of heat from small wires in air. Much had been published on that, but Dr. Langmuir found that the accepted laws were wrong; he discovered what they really are, and having determined that, and having found this effect of gas pressure on the rate of evaporation of tungsten, he put the two together and saw that it would be possible, with a large enough filament, concentrated by winding it in a helix, to produce a lamp considerably more efficient than any vacuum lamp. Before the first experimental lamp was life-tested, he had figured out on paper just about what life could be expected at a given efficiency from given sizes of lamp, both with nitrogen, and with argon of which there was not a bit available for use in this country at the time; and all the tens and hundreds of thousands of tests made since that day on different types of gas-filled lamps have in one way done little more than to establish over and over again the correctness of those calculations Dr. Langmuir had on paper before the first experimental lamp was life-tested.

Those two developments illustrate, I think, very well the distinction I wish to make, and I shall keep that distinction more or less before you in describing the work of our laboratory. Take the first kind of work, that is, the work which has a concrete object, a definite commercial goal, before it. Now that may be subdivided into two classes; that is, some of that work has to do with improving the existing products of the General Electric Co. Many lines of investigation of that kind are started by problems brought into the laboratory by engineers or by factory men. Some of these problems may require only a few days or a few weeks to solve. For instance, recently the switchboard department wished to find a material it could use in the bellows type of relay which was not only absolutely air-tight but would retain complete flexibility at very low temperatures. The material was found, animal bladder, and it was only a very short job to work out the technique for tanning it to render it fit for service. That is an illustration of the kind of problem which ordinarily can be solved in a few weeks.

On the other hand, some of those problems may require years of investigation. For instance, our engineers

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wished to find an insulating varnish capable of operating for a long time at relatively high temperatures without getting brittle. That meant an investigation of linseed-oil varnishes extending over a period of years, a study of the phenomena of the polymerization and the oxidation of varnishes, with the net result of producing some varnishes having the desired characteristics, materially improving the quality of the coils on which these varnishes are used, and incidentally, in many cases, reducing the factory cost. Some of these problems that are brought in, in that way, require a certain amount of productive work in the laboratory. For instance, some years ago, the railway department wanted a high-melting solder having certain characteristics, for brazing armature leads to commutator bars. We found an alloy of silver and cadmium that filled the bill, and ever since then we have been manufacturing in the laboratory that alloy in small quantities and supplying it to the railway department. Sometime the manufacture may be on a larger scale, arc-furnace work, for instance, where a non-magnetic alloy with high tensile-strength is required for high-frequency alternators.

Sometimes the problem is not one of new materials but of a new process, as, for instance, applying the copper brazing method developed in the laboratory to join certain points which before had been joined with rather expensive mechanical work and without an entirely satisfactory job. Still other investigations may be brought to us in the way of complaints from the outside. In some cases only an investigation may be required as to why, for instance, some particular turbine, in some particular locality, corrodes more rapidly than it should. On the other hand, it may mean that a considerable amount of work will be involved; as, for instance, the development of a new grade of carbon brush to take care of a particularly difficult motor service somewhere; but all of those kinds of work, brought to us in these different ways, some of them, of course, initiated by the laboratory, have to do, as I said in the beginning, with improving the existing apparatus of the company.

Researches with a Definite Goal

But there are other kinds of work, other investigations, which, like those I have just mentioned, have a perfectly definite object in view, but at the same time have as their purpose producing some new device and extending the scope of the work of the General Electric Co. Now, of course, the development of those new devices must in most cases be preceded either by some new observation or by a better understanding of some physical or chemical phenomenon. For instance, some years ago, while one of the gas-filled lamps was burning on life-test, it burned out near the lead, but it continued to burn, holding a short arc between the filament and the rather heavy lead. Enough current was passing through that arc and the rest of the filament in series to maintain the filament incandescent. It was a curiosity for several days, but we investigated the characteristics of that arc, an arc in gas between a hot filament and a relatively cold electrode; we found that such an arc has rectifying qualities; and the result was the development of the "tungar" rectifier, which is being made and sold in large quantities today for charging small storage-batteries. That was a new device resulting from a casual observation.

After Dr. Langmuir had discovered the laws governing pure electron-discharge in a high vacuum, Dr. Coolidge saw the possibility of building on those laws a new type of X-ray tube which would be much more reliable and constant in its characteristics, and much more convenient and easy to control, than the existing X-ray tube, and "the Coolidge tube," as it is known, was the result. Many of you may know more or less what that has done for X-ray practice. It not only gave the roentgenologist a much better tool to do the work he was already doing, but it has made possible many applications of X-rays which would not have been possible with the old type of tube, such as taking the X-ray to the bedside of a patient, or taking plates of the chest in from 0.025 to 0.100 sec., thus making stereoscopic work on the chest possible. We believe that the uses of X-rays are only just beginning, and it is the Coolidge tube which is opening up all these possibilities. So much for the kind of work that has a definite object in view.

We have at all times much of the other kind of work in progress in the laboratory. For instance, we have now one group of men who are passing homogeneous X-rays through little flasks of powder and photographing the result. They get spectra in that way, and from those spectra it is possible to calculate the relative positions and the relative spacings of the atoms which make up that particular material. In other words, it is one method of investigating the structure of matter, that is, the relation of the atoms to each other and their spacing. Another group is going farther than that, studying the structure of the atom itself, partly through studying the mechanism of the ionization of gases; and two or three other leads are being followed converging on the same point.

Those are only two instances out of a number of investigations of that kind which are being, or have been recently, conducted. Now it might well be asked by one unfamiliar with research, "What is the use of an industrial research laboratory doing that sort of work?" Part of the answer has already been given; that is, two things have been mentioned, the gas-filled lamp and the Coolidge X-ray tube, which owe their origin to exactly that sort of investigation. It has been our experience so far without exception that every important discovery which we have made regarding the fundamental laws or phenomena of physics or chemistry has resulted sooner or later in something of considerable practical utility and of commercial value to the General Electric Co.

As a matter of fact, I think it could be shown not only in the electrical art but in other arts besides, that many of the big steps forward, the radical things, have come from men who were not engaged directly in trying to improve that art. Take, for instance, the art of communication, the transmission of speech. The first telephone engineer was the man who made the first speaking-tube. He may have been a plumber or a tinsmith, and the plumber or tinsmith was the only telephone engineer for many years, but it was not a plumber or a tinsmith who produced the microphone and created real telephony. On the other hand, when the next big step was made, it was not the men who were working on telephony or upon telegraphy who discovered radio. Maxwell, Hertz and Lodge were physicists working to extend the limits of knowledge. It was they who discovered the waves which we now use in radio telegraphy. Now that radio telephony is beginning to overtake radio telegraphy, again we have a case in point; Dr. Langmuir, in the laboratory at Schenectady, was not thinking of radio, but was working on the "Edison effect" in incandescent lamps, when he made the fundamental discoveries on which he was able to base the design of a high-power vacuum-tube adapted to transmit radio, and in that way made possible,

for the first time, the transmission of speech by radio over long distances.

The academic study of the "Edison effect" gave us both the high-power transmitting tube for radio now known as the pilotron, and also the Coolidge tube; and we feel that that development is really only started. The tube which we had a year or two ago was good for a few hundred watts; now it is a few kilowatts, and we do not know where it is going to stop. I believe it would not be rash to prophesy that sooner or later high-power vacuum-tubes will make their influence felt in the real power field.

JUSTIFICATION OF INDUSTRIAL RESEARCH

For the reason, among others, that almost invariably from each new discovery or each new advance in science, pushing forward the boundary line between the known and the unknown, we do uncover something of commercial utility, we believe that an organization like our research laboratory is justified in continuing that kind of work. But there is another value to the laboratory, and through the laboratory to the General Electric Co., in that kind of work. We could not hold, in the laboratory, some of our best men unless we had that kind of work in progress. There is one type of mind that likes to see its ideas taking tangible form, likes to be working on something specific, so that it can chase that idea right through to something that it knows is of general utility; the man does his best work with a definite goal before him. There is another type of mind that is interested in the theoretical, in the abstract, rather than in the concrete, that does its best work and finds its greatest enjoyment in reaching after the unknown. Men of both types are necessary to our work. It is the men interested in the concrete, with a genius for embodying ideas in new and useful materials and devices, who for the most part produce the practical results that furnish the immediate financial justification for the maintenance of an industrial research laboratory; it is the men interested in theory, with a genius for discovering new and fundamental facts, who for the most part open up the broad new fields for exploitation.

So-called pure science and applied science are not antagonistic. Both are benefited by close association. Each is a stimulus for the other. Close contact with practical problems suggests continually new trains of thought to the scientific worker and helps him to keep his perspective. Close contact with research in pure science assists the worker on practical problems, by continually suggesting new methods of attack and new fields for development. In the industrial laboratory there is no sharp line between pure and applied science. One often merges into the other so gradually that no one could say where the line was crossed. A man who initially was engaged on an academic investigation may later find himself concentrating on the practical application of some phenomenon discovered in his original experiments.

It is because the industrial laboratory has brought together in mutual helpfulness pure science, applied science and engineering development, and has supplied them with adequate equipment backed by all the facilities of a large manufacturing plant, that it has achieved its marked success. It has proved itself a financial asset to industry and has contributed largely to the advancement of science. And great as the profits to industry have already been from the products of its laboratories, it is a safe prophecy that in the end industry will profit most by those results which today must be classified as contributions to pure science; for just as all of our engineering today is founded on the scentific discoveries of the past, so the engineering of the future will be based on the new knowledge science is now unfolding, and in that discovery a large and increasingly important part is being played by the industrial laboratory.

TWO-MAN ALTITUDE FLIGHT

WHAT is believed to be a new world's altitude record for pilot and passenger was made on May 6, by Lieut. J. A. Macready with Roy S. Langham as observer, when he reached a corrected indicated altitude of 34,150 ft. above sea level in a Lepere biplane equipped with a Moss supercharger. This airplane is the same one in which Major Schroeder made his world's altitude record for pilot alone and for pilot and passenger.

No difficulty was experienced by the pilot except the discomfort caused by the extreme cold at the great altitude and by the frosting of his goggles. Upon removing his glove to attempt to wipe ice from his goggles, his left hand became so stiff from cold that he lost the use of it until the warm air at a lesser altitude restored the circulation. The engine and supercharger functioned very satisfactorily.

While the figure given does not represent the true altitude above sea-level actually attained by the airplane, it is the figure upon which altitude records are granted. The corrected indicated altitude is dependent solely upon the pressure of the atmosphere in which the airplane is flying. If two airplanes attain the same corrected altitude on different days, which of the two airplanes reaches a higher true altitude depends on the temperature existing in the atmosphere between the airplane and the ground which is, of course, a matter of luck. It is for the purpose of eliminating the element of luck that the International Aeronautic Federation disregards true altitude in determining records, since it is obvious that, if one airplane reaches a higher corrected indicated altitude than another, which means lower air-pressure, it could out-climb the first airplane provided both were flown under identical atmospheric conditions. To obtain the correct indicated altitude, all the temperature and pressure corrections of the barograph used must be accurately known and carefully applied to the observed readings obtained on the flight.

Major Schroeder's corrected indicated altitude for the one-man record was 38,180 ft. by the Bureau of Standards computation, and for the two-men record 33,350 ft. computed by the Flight Test Branch of the Air Service at McCook Field. —Air Service News Letter.

The Requirements of Aeronautic Powerplant Development

By G. J. Mead[1] and L. E. Pierce[2]

Semi-Annual Meeting Paper *Illustrated with* Chart

THE object of this paper is to discuss the probable trend of aeronautical powerplant development, showing the reasons for the types which are likely to become more or less standard. Such a discussion involves a brief survey of the present situation, a review of the development of the various engine types, an analysis of the effect of their characteristics on airplane performance and a consideration of the proper installation of powerplants in the airplane.

The end of the war left us with very highly developed military powerplants, capable of enabling man to outdo the performance of birds in flight. This accomplishment was principally due to the necessities of the war. As a result of war requirements, designers strove increasingly to make lighter and more powerful engines. The limit of the power available from a single cylinder caused more and more cylinders to be added until, at the close of the war, engines were being developed with as many as 24. Various arrangements of cylinders existed. Practically all aimed at shortening the powerplants as much as possible. When more power was required and could not be secured by a given number of cylinders in a single bank, double, triple and even quadruple banks of cylinders were used, with a maximum of eight cylinders in a bank. Besides fixed-cylinder water-cooled engines, rotary air-cooled engines and radial air and water-cooled engines were used. Most of the engines were of what might be termed the racing type, being high-strung, light and delicate. The power range at the close of the war was from 60 to 1000 hp. Practically no commercial engines were developed. The war experience might well be likened to a tremendous experiment made under pressure, which came to a sudden stop with the ending of hostilities without answering the question of what were the best type of engine for each service. Therefore, a maze of engine types and powers are left, which are more or less fitted for particular military purposes but are, for the most part, unsuited for any other service.

The problem which now confronts the industry is one of establishing standard types for the powerplants required by each service, and setting up reasonable power requirements for each unit. With so many varieties of engines, it is obvivous that the art cannot be put upon a commercial basis, nor is the condition satisfactory for military purposes. In considering the situation from this point of view, it is necessary first to know what the services now are to which an engine can be put. There has been a small but increasing demand since the close of the war for what might be termed commercial engines which, due to the requirements of this particular work, have entirely different characteristics than the military powerplants previously available. It is still true that the principal demand is for military engines, and it is necessary for the safety of our country to develop this type unceasingly. There is great need for the development of satisfactory commercial powerplants. This is not because of the demand, but to stimulate it. With these engines available, the success of the first undertakings in commercial aviation will be assured. A third service, which might be termed sport, will be developed ultimately. This service will require engines as different from the others as our passenger-car engines are from those used for racing or trucks.

For each service, factors must be developed to permit the making of correct comparisons of the performance of the different engines. The engineering method of doing this consists in comparing certain constants for each design, which have been developed by experience. These constants necessarily apply only to a particular set of requirements or, in other words, to one particular service. It so happens that we are accustomed now to judge the performances of all aeronautical engines by constants which are in reality only applicable to military engines. For this reason, aeronautical engineers unconsciously consider that an engine must weigh less than 2 lb. per hp., and that the fuel consumption is satisfactory if it is in the neighborhood of 0.50 lb. per hp-hr. Such an engine rarely lasts more than 50 hr. without a top overhaul, which is comparable with 6000 miles of air-travel. The airplane designer is inclined to believe he must have maximum power and minimum weight at any cost. In this case, the cost is the reduced life of the engine. The commercial requirements involve the use of entirely new factors, since the prime requisites of commercial work are efficiency and durability, which changes the engine design considerably. Our definition of durability is long life. All aeronautical engines must be reliable, but in the past durability has been sacrificed for light weight per horsepower. The weight factor for engines with these requirements becomes at least 3½ lb. per hp. Commercial engines should run at least 16,000 miles without a top overhaul, with a fuel consumption approaching 0.40 lb. per hp-hr. These constants contrast sharply with those for the military requirements. In this connection it is interesting to compare these figures with those for the Wrights' first engine, which weighed approximately 7.6 lb. per hp. This is a decided indication of the progress that has been made.

Consideration is given next to the different types of engine, and then a study is made of the particular types most suited for each service. Before taking this up, it is perhaps advisable to point out that fuel consumption is not related to the type of engine. By using high compression and sacrificing some power, which means increasing the weight constant, approximately equal fuel

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economies can be obtained from any of the various types of engine. It follows that military engines naturally sacrifice economy to keep the weight factor low. The form or type of an engine is due principally to the power limitations of an individual cylinder, as previously pointed out. From the standpoint of the airplane designer it is desirable to keep the engine as compact as possible, principally limiting those dimensions which increase head-resistance and tend toward bulky powerplants. This encourages the use of engines having a single vertical bank of cylinders. When more power is required than can be obtained with this arrangement, it is necessary to go to multiple-bank types. It is obvious, therefore, that it is necessary to take up next the question of what limits the power of an individual cylinder.

The power available from a single cylinder depends upon the dimensions of the cylinder, the speed of operation and the compression ratio. Other items, such as the fuels used, the timing and the size of the valves, will not be considered in this connection, since we are able so far to provide for these requirements regardless of the size of the cylinders and they therefore do not impose any limits on cylinder performance. Taking up the cylinder dimensions first, we find that a 5½-in. bore cylinder was practically the limit at the close of the war, which made available some 45 hp. per cylinder and resulted in a large number of cylinders where considerable power was required. The limiting factor for the bore appears to be the reciprocating parts. With large bores, the pistons become difficult to cool in the usual manner. This is not an insurmountable restriction, since several expedients are known for cooling pistons in other ways. The principal reason is that the power required to overcome the inertia of the reciprocating parts increases rapidly with the increase in the cylinder bore at a given speed and, since this power must be subtracted from the available indicated horsepower of a given cylinder, the brake horsepower is reduced. Indicated mean effective pressure is independent of cylinder size, at least so far as our experience to date goes. Therefore, as the cylinder bore increases, the reciprocating weights increase and the speed of the crankshaft must be decreased to permit the obtaining of good brake mean effective pressure. At present, cylinders of 7-in. bore are being operated successfully at speeds as high as 1400 r.p.m., with brake mean effective pressures corresponding to those obtained with any of the smaller cylinders. Powers as high as 65 hp. are now obtainable from a single cylinder of this size. This is approximately a 30-per cent increase in power over previous practice, and permits important changes in the types of engine for the different powers required. Carrying bore sizes to an extreme, a point will be reached where no increase in power will be obtained by further increases, due to the very low crankshaft speed which the reciprocating parts of this particular size will permit. What this point is, has yet to be proved.

Taking up the question of stroke, we find that the principal restriction upon increasing it beyond a certain dimension similar to the bore, is the rapidity with which the weight per horsepower of the engine increases. It has been found that a "square" engine, one having the bore equal to the stroke, makes the lightest possible construction. When the valves are placed in the head, it is very difficult with long strokes to secure sufficient valve-area, and still cool them properly so that they will stand up under the temperatures resulting from the high mean effective pressures. Of course, increasing the stroke increases the piston speed at any given number of revolutions per minute of the crankshaft, but so far no difficulties have been experienced from this cause. It appears, therefore, that some increase in power can be had by lengthening the stroke on commercial engines, but not for military engines owing to the weight limitations.

The speed of the engine, or its rate of doing work, is an important factor. Of course, in obtaining maximum power with a minimum weight it is necessary to run the engine just as fast as possible. The limiting feature here becomes the propeller speed. With the pursuit ships it is possible to run the propeller up to 2100 r.p.m., but for commercial work, due to lower flying speeds, it is not desirable to turn the propeller nearly so fast, because of the effect on propeller efficiency. A propeller speed of 1800 to 2000 r.p.m. is considered more or less standard for military work, and from 1000 to 1400 r.p.m. for commercial work. In the case of a dirigible, the propellers turn about 550 r.p.m. The question immediately arises, Why not gear the engine? For military work, no commercially manufacturable gearing has been devised which permits a reasonable powerplant weight per horsepower. It seems perfectly possible to use gearing for very large commercial airplanes and dirigibles, where the size and weight of the gear-reduction is not of such importance. It has been found desirable to increase the compression-ratios as much as possible, to permit the maximum power output of the engines at great altitudes. The maximum compression-ratio from which a gain in power can be secured is 7 to 1. So far, the size of the cylinder does not seem to limit the use of such a compression. Above 5¼ to 1 it is desirable to use some anti-knock substance such as benzol, if it is necessary to operate the engine on the ground under full power.

From the foregoing, it will be seen that we have not reached, as yet, the maximum power available from a single cylinder. It is true that, as the bore increases, the speed must be decreased somewhat. It is desirable to keep the ratio of bore and stroke as nearly unity as is possible, especially for military engines. The maximum compression-ratio is fixed at 7 to 1. Without dope, it is possible to secure 130 lb. per sq. in. mean effective pressure in water-cooled cylinders, and with dope as high as 145 lb. per sq. in. it follows that we shall be able to produce much larger powerplants with fewer cylinders than formerly, if a demand for them arises.

Future Types of Engine

In taking up the consideration of future types, based on the above conclusions, it is desirable to dwell for a moment upon a few of the general conditions and requirements facing the aeronautical engineer today. All types of engine must operate under wide variations of temperature and pressure, the ranges of temperature being from 50 deg. below to 100 deg. above zero fahr. and of pressures from the usual sea level of 30 in. of mercury to as low as 8 in. Besides these atmospheric conditions, aeronautical work requires that the engines be the lightest of all prime-movers, which involves the use of the strongest and lightest materials known. Reliability is an absolute necessity since, in heavier-than-air machines, the failure of the powerplant deprives the pilot of the ability to navigate his ship at will. This does not mean necessarily that any serious accident need result, but it prevents accurate maneuvering.

Due to the variations in the air temperature, it becomes more difficult to provide means for keeping the engines at the proper operating temperatures. Both air

and water-cooling have been used, but no standard has been determined upon, except that water-cooling seems to be the more generally favored. Great strides are being made in the development of air-cooled cylinders for aviation work. The British report being able to obtain as high mean effective pressures as with water-cooled cylinders. So far, the maximum mean effective pressure which has been developed in this country with large air-cooled cylinders is in the neighborhood of 110 lb. per sq. in. With relatively small cylinders, pressures as high as 125 lb. per sq. in. have been secured. Thus far satisfactory air-cooling has been limited to radial engines. Air-cooling has advantages for an airplane, due to the tremendous differences in temperature under which the engines are forced to operate. It also makes the engine somewhat less vulnerable when used for military purposes. This does not seem to be a very important feature, since most planes that are shot down apparently fall because of fire rather than any other cause. The disadvantages of this type of cooling are that it allows, so far, relatively low mean effective pressures, increases the parasite resistance of the airplane considerably, and does not permit easy temperature-control. Another point that must be borne in mind is the effect of the accessory weights upon the ultimate powerplant weight. For example, let us assume that the maximum power that one carbureter can take care of is 200 hp. If 300 hp. is required, two carbureters will be necessary, but without additional carbureter weight it may be possible for them to supply satisfactorily a 400-hp. engine. It will always be found, therefore, when it is necessary to double up on accessories such as magnetos and carbureters, that unless the power is also proportionally increased the weight of the resulting engine will be in excess of that in which the accessories are used to their maximum capacity. With the general restrictions in mind, we are able to proceed with a detailed consideration of the different types of engines in connection with the cylinder arrangements.

The "in-line" type of engine is most particularly adapted to aviation work, due to its low head-resistance and the simplicity of its parts. For this reason it makes an ideal commercial type of engine. Its weight, however, is greater per horsepower than that of the V-type engine. Four cylinders in line are not satisfactory, due to both the power limitations and the vibration. Six cylinders form an ideal combination, since it is possible to obtain powers as high as 400 hp. from this number, and the engines are perfectly balanced mechanically. The weight factor for a six-cylinder engine is from 2.5 to 3.5 lb. per hp., depending upon the service. The limit of the number of cylinders that can be put satisfactorily in one line is eight, due to the torsional vibrations of the crankshaft. The weight of such an engine is about 20 per cent greater per horsepower than for a six-cylinder engine, due to the long crankshaft and crankcase, but for such uses as dirigible work the durability and efficiency of this type are sure to recommend it. Over 500 hp. is available in a single unit of this type. Six-cylinder engines are much used, but "in-line-eights" have not as yet come into service.

Under V-type engines are classed the lightest powerplants; therefore, it is a type much used for military service. The smallest number of cylinders in a single bank that has been used successfully is four. In this case, the two banks are 90 deg. apart. The V-type eight-cylinder engine is undoubtedly the lightest water-cooled engine known, for the reason that it has the shortest crankshaft for a given power. Weight factors of 1.45 lb. per hp. have been secured from this type of engine. For pursuit work this is the most important consideration. It makes the use of these engines for this work practically a necessity, especially when consideration is given to the effect of weight upon the performance of an airplane, which point will be taken up later. The one drawback to the V-type eight-cylinder construction is the tendency for the engine to vibrate in a horizontal plane. This is noticeable only with heavy reciprocating parts and at slow engine speeds. It has already been proved that eight cylinders of sufficient power to give 400 hp. at 2000 r.p.m., which is a pursuit propeller-speed, do not involve reciprocating parts of sufficient weight to make the vibration serious to the plane structure. It is necessary, of course, to exercise care in the design of mountings for these engines to take care of the horizontal vibration. Pursuit work is a matter of performance, just as with a racing automobile. In neither service is exceptional smoothness required if it in any way deducts from the performance. V-type 12-cylinder engines are used considerably for short-distance bombing airplanes, where the engine weight is a factor. Small engines of this type will undoubtedly be in demand for pleasure purposes, just as they are in automobile work. The best weight-factor from a 12-cylinder engine is in the neighborhood of 1.9 lb. per hp. With a given displacement the 12-cylinder engine presents much more exposed cylinder and port area to the cooling water and therefore a considerably larger radiator is required than for a fewer number of cylinders. This increases the total weight of the powerplant and also its head-resistance.

Radial Engines

Serious consideration has been given to radial engines, particularly by the British. It was originally thought that this type of engine would be lighter per horsepower than any other and that, with its concentrated mass it would be particularly adaptable to pursuit work; and in addition that, the customary cooling system being eliminated by using air-cooled cylinders, the plane could not be brought down due to any damage to the cooling system. It has been found that the weight-factor for this type of engine is about 2 lb. per hp., when the design is such as to permit the engine to operate satisfactorily for 50 hr. This is practically the same weight-factor as for the V-type eight-cylinder water-cooled engine complete with water and radiator, so that there is really no saving in weight. Moreover, a radial engine of a similar power to that of the V-type eight-cylinder engine has an enormously increased parasite resistance. Radial engines properly balanced are very smooth-running and compact as regards the concentration of weight. The temperature control is very difficult. Fuel consumptions similar to those obtained with water-cooled engines are now possible, but the oil consumptions are just about twice those for the standard water-cooled types. Water-cooled radial engines have been built, such as the Salmson, made in France, but there does not seem to be any particular value in such designs.

The rotary type of engine was much used before the war, but was rapidly discarded with the increasing demands for power, since the crankshaft speed of such an engine is limited by centrifugal force. Therefore, it was necessary to use extremely large cylinders to obtain even reasonably high powers at the restricted operating speed. Small rotary engines were limited to from 1200 to 1400 r.p.m. This type of engine has now practically disappeared. Other arrangements of multi-cylinder engines

have been used, such as a so-called W-type with 12 and 18 cylinders, but the weights and widths of such engines do not warrant their use except for large powers in military work, and their complication tends to preclude their becoming popular for commercial work.

It seems almost advisable to restrict the number of cylinders to 12, as a maximum. If more power is required than can be secured from this number in a single unit, more units should be added. The situation is becoming practically identical with that faced by marine engineers, and we find that they have concluded it is more satisfactory to use several power units, rather than tie up their entire power in a single unit. It may be very alluring to the plane designer to consider 2000 hp. in a single unit, but two 1000-hp. units are very much more likely to bring the ship home. Several power units can be utilized for very large commercial ships, if centrally located in power nacelles. In this case they can be arranged to drive separate propellers, or connected together in such a way as to deliver their power to a common shaft. In the case of dirigible work the former arrangement is preferred. The problems involved in working out the latter arrangement are yet unsolved, the principal difficulty being that a propeller tends to turn as a flywheel at a constant velocity, whereas the various engines connected to it through the gears do not synchronize, and thus load the gearing intermittingly with disastrous results. Considerably fewer problems are involved if each engine has its own gearing and its own propeller.

Variations in Engine Characteristics

It is desired to give definite values to certain engine characteristics and to vary these values between limits, to determine, primarily, what the effect will be on the performance of an airplane. In such a discussion it is necessary to consider overall dimensions, engine weight per horsepower, fuel economy, cooling economy and altitude performance of the engine as factors. In general, overall dimensions will affect the maneuverability of the airplane, the parasite drag of the fuselage group and vision. These factors apply particularly to airplane designs having a single engine installed in the fuselage, for either military or commercial use.

The tendency of engine design toward radial or multiple-bank types is governed to a large extent by the desire to furnish short-coupled compact units. The application of such engines to airplane design has as its object increasing the maneuvering ability of the airplane. This question arises particularly in reference to engines for pursuit planes, and in the past has received much attention in the new development of such types of engine. It is true, all other things being equal, that the airplane having the greatest concentration of mass will prove to be the most maneuverable; but this consideration is only one in the very complex study of airplane stability. It will be remembered that maneuverability and stability are related, in that maneuverability will be most pronounced under conditions of neutral stability. If the airplane is designed to be highly stable, it will be stiff and will strongly resist rapid changes in flight attitudes, which is really a measure of its maneuvering ability. Concerning the airplane alone, the size and position of the lifting and control surfaces, the proportions and distribution of keel surfaces, the balance of the airplane and perhaps also the influence of different speed ranges on the airfoil surfaces, all have their bearing on the resulting ability of the airplane to maneuver. The effect due to mass concentration as influenced by engine dimensions, within reasonable limits, is probably the least important of these several factors. To bear out these assertions we can refer to that rather hackneyed example of certain of the German pursuit airplanes which were designed around a heavy vertical six-cylinder type of engine and yet possessed to a remarkable degree speed in maneuvering and readiness of control. These features were undoubtedly due to the aerodynamic characteristics the designers were able to derive from the choice and disposition of their lifting and control surfaces. There is much need of further experimental research in aerodynamics to develop practical data that can be put at the disposal of the practising airplane designer to assist him in the positive development of his design to a prescribed condition of stability.

We can consider properly the influence of the engine on the drag of the fuselage as increasing that drag when engine dimensions protrude beyond the usual fuselage lines. To complete this factor, the resistance offered by the cooling element should be considered also. For the water-cooled engine we must add the drag of the radiator. For the air-cooled engine we must allow for the high drag caused by protruding cylinders. An example is chosen of a design having about 375 hp. Two water-cooled engines are considered, one of which fairs readily into the fuselage lines, the other has protruding cylinder-blocks. One air-cooled engine is considered. The assignment of a proper drag coefficient is very difficult in this latter case due to interference to air-flow between cylinders. The total frontal area has been proportioned arbitrarily between cylinder shapes of high unit-drag and fuselage shape of minimum unit-drag. It is believed that the proportions chosen probably favor the design. The following relations are based on the usual formula:

$$D = D_c A V^2$$

In which

D = the drag
D_c = the pressure in pounds per square foot per miles per hour
A = the area in square feet
V = the speed in miles per hour

The values of $D_c A$ are determined, as they are directly proportional to the drag, and are shown in Table 1

TABLE 1—VALUES OF $D_c A$

Parts	Values of D_c	Values of A	Average $D_c A$	Best $D_c A$
Water-Cooled Engine				
Best fuselage	0.0006	10.75	....	0.00645
Average fuselage	0.0008	9.25	0.0074	....
Radiator, 9-in. core depth	0.0014	2.00	0.0028	0.00280
Total for fuselage group			0.0102	0.00925
Air-Cooled Engine				
Cylinders	0.0030	4.00	0.0120	....
Fuselage	0.0004	5.25	0.0021	....
Total for fuselage group			0.0141	

From these data we can determine that for water-cooled engines the drag of the fuselage group may be increased by about 10 per cent when the engine shapes are poorly adapted to usual fuselage lines. For a large radial engine this value may be as high as 52 per cent. The drag of the fuselage group for pursuit types of airplane will be 30 per cent of the total drag. This means that the total drag of the airplane will be increased, considering best water-cooled engine shapes as a base, about 3 per cent for poorer shapes of water-cooled engine and 16 per

cent for air-cooled engines. This increase in total drag will reduce the fineness of the resulting design, and so reduce the high speed. The effect will be of importance in military types, because of the tactical value of maximum speed. It will be important in commercial designs also, because of its effect on mileage obtained for a given fuel-consumption. The figures noted for the air-cooled design indicate a loss in high speed of 5 per cent, compared to best water-cooled design, which is an appreciable amount. The term "fineness" refers to the aerodynamic efficiency of the airplane; it is a function of L/D, the lift-drift ratio.

The subject of vision will be passed over briefly as it is very difficult to isolate the effects of engine size alone on a given vision. The design of the exhaust-manifold plays as conspicuous a part in obstructing vision as do cylinder-block positions. Furthermore, the engine is located at a point which is naturally blind because of the presence of the fuselage, and an increase in the blind area due to added engine interference will be comparatively small. This is rather remarkably shown in designs around the larger types of radial engines, such as the 375-hp. size, where a careful positioning of the pilot will result in very little added interference to vision. Cylinders Nos. 2, 3, 8 and 9 will be obstructing vision to some extent but their interference is of minor importance.

Reviewing briefly, we have pointed out that overall dimensions affect airplane maneuverability but that, within reasonable limits, their importance is probably secondary to other factors relating more directly to aerodynamic design. Extreme maneuverability is, in any event, a specialized characteristic required primarily of military pursuit airplanes. Overall dimensions, and particularly the form of engine design, as to whether it is water or air cooled, will influence appreciably the high

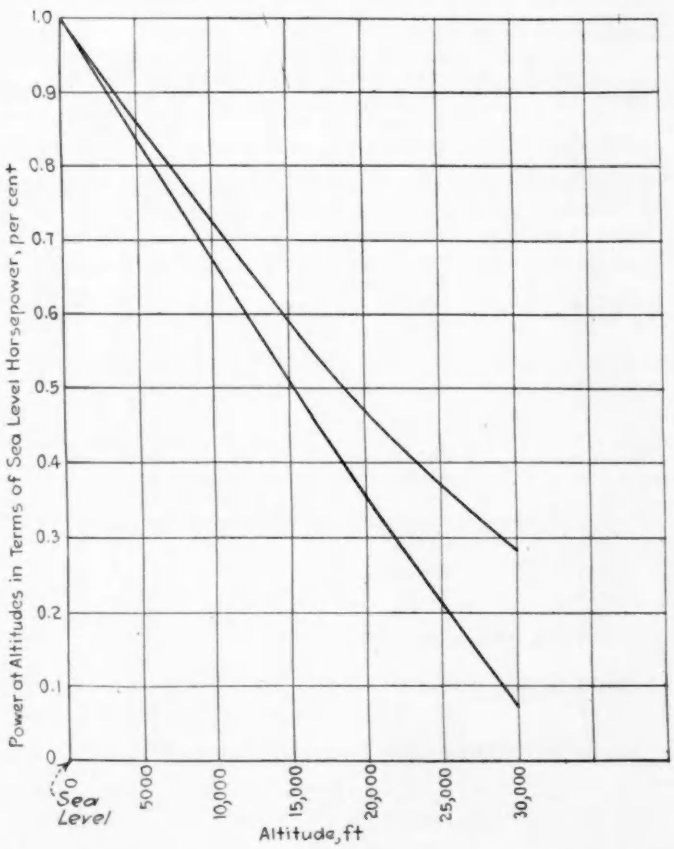

FIG. 1—VARIATION OF ENGINE HORSEPOWER WITH ALTITUDE AT FULL THROTTLE

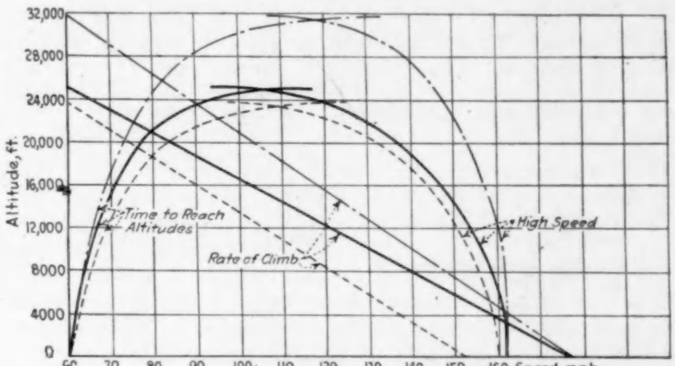

FIG. 2—RELATIVE PERFORMANCE OF TWO AIRPLANES IN WHICH THE ENGINE WEIGHT, COOLING ECONOMY AND THE PERFORMANCE OF ENGINE AT ALTITUDE VARY

speed obtainable for a given design. The importance of this characteristic is both military, for tactical reasons, and commercial, for reasons of operating efficiency. Overall dimensions have less effect on vision than usually is thought to be the case. Vision is governed not only by the extent of the obstructing bodies but by the positioning of the pilot with respect to these bodies.

The factors of engine weight per horsepower, fuel economy, cooling efficiency and engine performance at altitude, will be considered together as, in each case, their influence is similar as affecting the general flying performance of an airplane. In questions involving powerplant weight, it will be impossible to cover in detail the full range of ratios of this weight to the total weight as given by various sizes of airplane. It is necessary to tie down to one certain point in this range, which has been done in the examples chosen to illustrate these points.

Fuel economy and its effect on the total weight of the fuel carried has more bearing on long-flight duration than engine weight. We can assign the limits for these two factors as being engine weight and fuel economy without further detail and with a reasonable degree of accuracy. Assume that the engine weight will be increased 1 lb. per hp. and that there will be a reduction in fuel economy of 0.1 lb. per hp-hr. With such values it will be seen that a flight of 10 hr. is required to make fuel economy of greater importance than engine weight. This example, although crude, has its value as indicating rather clearly that engine weight will always be the important factor in airplane design, but that fuel economy will usually be of more importance in airship design. This, of course, assumes that the airship will be the medium of transportation for air-travel covering great distances. An exception may be found in certain special heavy-duty long-distance bombing or transport airplanes. The commercial airplane will usually find it much more advantageous to make relatively short hops, thus allowing the proportion of pay-load to fuel-load to be as high as possible.

To indicate the effect of engine weight per horsepower and cooling economy on airplane performance resort is made to hypothetical airplanes A and B and the result is shown in Table 2. Airplane A is taken as a standard. Airplane B differs from airplane A only in the use of an engine of greater weight per horsepower and one which will require ⅓ more radiator cooling capacity than the engine for airplane A. The weights for powerplant are bulked and denoted by X. The weights of the structure complete, including the pilot, controls, instruments and the like, are bulked and denoted by Y, which equals 0.892

X. The value of Y in terms of X has been determined by obtaining the average figures given by several accomplished designs. The value of Y for airplane B, 1277 lb., is not 0.892 X, as airplane B is geometrically identical with airplane A. An increase in weight has been allowed which is assumed to hold the structural strength of the two airplanes the same. The characteristics on which the performance of airplanes A and B are dependent are shown in Table 3. The reduction in "fineness" has been based on the reduction in the L/D-ratio for the airplane, as affected by the increase in drag due to the increase in radiator size required.

TABLE 2—HYPOTHETICAL AIRPLANES COMPARED

Inclusive of	Weight of Airplane A, lb.	Weight of Airplane B, lb.
Engine, Engine Water, Radiator and Water, and Propeller	840 (2.1 per hp.)	1,040 (2.6 per hp.)
Gasoline, Lubricating Oil and Tanks	503	503
X	1,343	1,543
Y	1,192	1,277
Total Load	2,535	2,820

TABLE 3—CHARACTERISTICS ON WHICH PERFORMANCE DEPENDS

Characteristics	Airplane A	Airplane B	Airplane C
Total Weight, lb.	2,535	2,820	2,535
Area, sq. ft.	280	280	280
Engine Power, hp.	400	400	400
Weight, lb. per sq. ft.	9.05	10.08	9.05
Weight, lb. per hp.	6.34	7.05	6.34
Fineness, deg.	115	113	115

Before developing these performances we will consider the effects of engine performance at altitude. Airplane A is again the standard. Airplane C is powered by an engine of weight per horsepower and cooling economy similar to those for airplane A, but its altitude performance is according to the data given in Table 4, and Fig. 1, under engine No. 2. The characteristics on which the performance of airplane C is dependent are given in Table 3. The engine altitude-performance for airplanes A and B is according to the data given under engine No. 1, taken from two actual engine performances. The results of the performances developed for airplanes A, B and C, are given in Table 5 and are shown graphically by the curves of Fig. 2.

This study, see also Figs. 1 and 2, derives a means for measuring the effects on the complete performance of the airplane of increased weight and of increased drag, as governed by the engine design, and of better altitude performance of the engine. By thus increasing the total weight of the airplane 11 per cent and by decreasing its L/D-ratio 3 per cent

TABLE 4—EFFECTS OF ENGINE PERFORMANCE AT ALTITUDE

Altitude, ft.	Engine No. 1, per cent	Engine No. 2, per cent
Sea Level	100.0	100.0
5,000	83.4	85.8
10,000	67.7	72.1
15,000	50.8	59.1
20,000	34.9	47.1
25,000	21.5	37.5
30,000	7.6	28.5

(1) The rate of climb is reduced by values of from 22 per cent at sea level to 40 per cent at a 20,000-ft. altitude

(2) The time to climb is increased by values of from 29 per cent at 5000 ft. to 30 per cent at a 20,000-ft. altitude

TABLE 5—RESULTS OF PERFORMANCES DEVELOPED

	Airplane A	Airplane B	Airplane C	B Performance Reduced to Percentage of A Performance	C Performance Increased to Percentage of A Performance
Weight, lb. per sq. ft.	9.05	10.08	9.05		
Weight, lb. per hp.	6.34	7.05	6.34		
Fineness, deg.	115	113	115		
Engine	No. 1	No. 1	No. 2		
Speeds, m.p.h, at altitude, ft.:					
Sea level	162	160	162	98.70	100.0
5,000	161	158	161	98.70	100.0
10,000	156	153	161	98.00	103.0
15,000	149	146	158	98.00	106.0
20,000	136	131	153	95.00	112.5
25,000	103	113 (23,000 ft.)	145	.....	141.0
30,000	...	...	131	.....	.....
Absolute Ceiling	94	98	107	.....	.....
Ceilings, ft.:					
Service	24,000	22,400	30,400	93.25	126.8
Absolute	25,150	23,750	31,800	94.50	126.5
Rate of Climb, ft. per min.:					
Sea level	2,340	1,840	2,340	78.50	100.0
5,000	1,870	1,450	1,970	79.10	105.3
10,000	1,410	1,060	1,600	75.70	113.5
15,000	940	670	1,210	72.00	128.8
20,000	480	290	860	61.70	179.2
25,000	...	...	500	.....	.....
Time to climb, min.:					
5,000	2.38	3.07	2.33	129.00	97.8
10,000	5.45	7.07	5.14	129.70	94.3
15,000	9.75	12.90	8.66	132.40	88.7
20,000	17.01	23.84	13.49	140.00	79.2
22,400 Service Ceiling of B	.....	37.05	.....	......	....
24,000 Service Ceiling of A	33.15	.....	.....	......	....
25,000	.....	.....	20.92	......	....
30,000	.....	.....	39.03	......	....
30,400 Service Ceiling of C	.....	.....	44.55	......	....

(3) The ceiling is reduced by about 6 per cent
(4) The high speed is reduced by values of from 1 to 5 per cent. This loss in high speed is due almost entirely, however, to the reduced fineness of the airplane

By varying the relative percentages of power output at altitude from equal values at sea level to a relative increase of 375 per cent at 30,000 ft.

(1) The high speed is increased by values of from 0 per cent at sea level to 13 per cent at a 25,000-ft. altitude
(2) The ceiling is increased by 27 per cent
(3) The rate of climb is increased by values of from 0 per cent at sea level to 79 per cent at a 20,000-ft. altitude
(4) The time to climb is reduced by values of from 2 per cent at 5000 ft. to 21 per cent at a 20,000-ft. altitude

The value to military airplanes of these increases in performance are unquestionable, especially as they relate to climbing ability and ceiling because they are the more pronounced. The tactical advantage which the fighting pilot will have in an ability to out-climb and out-reach his opponent is well understood. Commercial types of airplane will benefit most in an ability to substitute for unnecessary dead-load or dry weight, the same weight in the form of pay-load. The value of engine efficiency for commercial work will be particularly great in the ability it gives to maintain higher speeds at cruising altitudes, and so increase mileage for a given weight of powerplant and fuel consumption.

We have shown what increases in airplane performance can be expected through variations in engine characteristics from average to best values. If still greater increases are required, either to put military aviation on a higher plane or to permit commercial aviation to be established on a sound economic basis, we cannot expect to derive such increases simply by further refining of present types of design, but rather we must develop other forms of design, perhaps radical in nature, which will accomplish the work of flying more efficiently than our present types of apparatus can.

Characteristics for a High-Speed Airplane

Before leaving the subject of engine and airplane characteristics, it will be of interest to make an estimate of what these characteristics must be for a very high-speed airplane. Popular interest centers around the 200-m.p.h. mark. We often speak as though we can attain that speed simply because we desire to do so. It is well to realize just what the requirements are in order that an airplane, designed to our best usual practice, carr fly 200 m.p.h. in bona-fide level flight. The results of our last Pultizer Trophy Race serve to give us a basis on which to build an estimate. The characteristics of the actual airplane are given in Table 6.

TABLE 6—CHARACTERISTICS OF THE ACTUAL AIRPLANE

Weight, lb.	1,875
Wing Area, sq. ft.	250.5
Engine Power, hp.	367
Assumed Propeller Efficiency, per cent	80
Weight, lb. per sq. ft.	7.50
Weight, lb. per hp.	5.10
High Speed, m.p.h.	170.25
Wing Section, R.A.F.	No. 15

A = Wing Area, sq. ft.
V = High Speed, m.p.h.
$L_c = 1875 \div [250.5 \times (170.25)^2] = 0.000259$
$L = L_c A V^2$
$i = -0.2$ deg. = corresponding angle of incidence
$D_c = 0.000032$
$D_w = D_c A V^2 = 0.000032 \times 250.5 \times (170.25)^2 = 232$ lb. = Wing Drag
$D = (367 \times 0.80 \times 375) \div 170.25 = 647$ lb. = Total Drag
$\frac{DV}{375}$ = Engine horsepower times propeller efficiency
$D - D_w = 415$ lb. = Parasite Drag
$(D_c A)_p = 415 \div (170.25)^2 = 0.01432$ = Parasite Drag Unit
$D_p = (D_c A)_p V^2$

The actual airplane was of standard size and wing-loading and did not represent suitable wing-area, fuselage dimensions or chassis size for a light racing airplane. Therefore certain reductions in size are assumed which are to be expected in a design developed solely for speed purposes. Assuming that the weight loaded is 1740 lb., taking account of the reduced dimensions throughout, and that the landing speed shall not exceed 65 m.p.h., the maximum L_c for the wing is 0.0030 lb. per sq. ft. per m.p.h. Then,

$A = 1740 \div [0.0030 \times (65)^2] = 137$ sq. ft.
$L = L_c A V^2$
$L_c = 1740 \div [137 \times (200)^2] = 0.00318$ lb. per sq. ft. per m.p.h. at high speed
$i = +0.1$ deg. = corresponding angle of incidence
$D_c = 0.000033$
$D_w = 0.000033 \times 137 \times (200)^2 = 180.8$ lb.

For this type of airplane, the distribution of drag at an angle of 0 deg. is of the order shown in Table 7.

TABLE 7—DISTRIBUTION OF DRAG, $i = 0$ DEG.

	Standard Design	Drag Percentage	Special Design	Reduction Factor	New Percentage
Wings, lb. per sq. ft.	7.5	30.0	12.7	....	....
Wind Bracing	2-Bay	7.2	1-Bay	0.60	4.32
Fuselage, sq. ft.	11.5	27.7	8.63	0.75	20.78
Chassis	....	22.6	....	0.75	16.95
Tail Surfaces	....	12.5	No reduction	1.00	12.50
Percentage of Drag which is Parasitic		70.0	....	....	54.55

Under such conditions the parasite-drag unit will be reduced to

54.55/70 = 0.78 of its original value
New $(D_c A)_p = 0.01432 \times 0.78 = 0.01118$
$(D_c A)_p V^2 = 0.1118 \times (200)^2$
= 447.2 lb. = Parasite Drag
$D_w = 180.8$ lb.
$D_p + D_w = 628$ lb. = Total Drag
Required Horsepower = $(628 \times 200) \div (375 \times 0.8)$ = 418 hp.

Weight, lb.	1,740
Area, sq. ft.	137
Engine Power, hp.	418
Weight, lb. per sq. ft.	12.70
Weight, lb. per hp.	4.15
High Speed, m.p.h.	200

One can well imagine the care required in the design of both engine and airplane to meet this power, weight and fineness.

The matter of the value of retracting the chassis immediately presents itself. A study, similar to the one already presented, has been carried through on the basis of eliminating completely the chassis resistance. The assumptions that were made and which are considered to be fair are (*a*) an addition of 60 lb. of weight, to cover the retracting mechanism complete, and (*b*) that the fuselage dimensions are not altered to provide suitable space for the retracted chassis. Under these condi-

tions the required power becomes 384 hp., which is more definitely within the range of possible design limits.

Powerplant Installation

Cooperation between the builder of engines and the constructor of airplanes is the most essential factor required in the development of successful and serviceable installations of powerplants. The aircraft industry is at present divided into these two well-defined activities. Each has its own distinct engineering problems and, naturally, the viewpoint of one is not identical with that of the other when considering the requirements of the industry as a whole. However, the interest and advantage of both are reasons for finding common ground in solving the engineering problems incidental to the installation of the powerplant. It is essential that the airplane and the engine shall be assembled into the powered airplane with due consideration given by both parties to the many complex problems involved.

The airplane designer should consider the detailed requirements of the engine he is using as soon as he makes his first study of the general arrangement of his design. Simplicity in the design of engine systems is in itself an assurance of their dependability, and a simple design can be effected most readily by grouping all items relating to the engine as near to it as is practicable, without interfering with accessibility. The necessary cooperation between engine and airplane designers must be asserted during the design periods when all controlling factors can be understood thoroughly and considered, and suitable compromises reached. It is well known that a completed design, if improperly executed, is very difficult to change and correct. When the necessary changes are undertaken they are always costly and often ineffective, due to many other limiting factors.

Before developing designs of powerplant installations it is necessary to know what the exact capacities required by the several systems are, and these capacities must be provided if the engine is to operate at maximum power. It is the function of the engine builder to know what the particular installation requirements for his engines are. He must be prepared to recommend suitable types and capacities for all accessory equipment. Recommendations of this sort will be governed largely by the type and characteristics of the airplane involved, and general specifications covering any and all cases cannot be written satisfactorily. For this reason each design must receive its special study before accurate and most applicable recommendations can be offered. This study must necessarily be gone into jointly by the airplane and the engine builders. The latter will probably supply a complete powerplant eventually rather than an engine which can be installed as a unit in a given airplane design. When airplane sizes have increased sufficiently to allow complete engine nacelle designs, such a procedure can be wholly practical.

Propellers, although their design is based largely on aerodynamic calculations, belong primarily to the powerplant group. The development of propellers for military airplanes has been dependent largely on the ultimate performance desired of these airplanes. The engine to be used affected the propeller design principally in determining the speed at which the propeller should turn and the maximum power it would need to absorb at that speed. Economy in engine operation is very greatly dependent upon the values of running speed and developed power for that speed. To realize greatest economies in engine operation, propeller designers must, in the case of types of airplane in which operating economy is paramount, consider the ability of the engine throughout its full power range, rather than only its maximum performance.

A most important consideration in the design of powerplant installations is that of engine supports or mounting. It is not only necessary that the mounting provide a suitable support for the engine, but its design must also be arranged to permit accessibility to the essential parts of the engine, such as pump connections, strainers, relief valves and magnetos. This will require a consideration of these points when determining the position of the truss members or the type of design. For instance, the use of a veneer skin at the engine section, which is designed to carry shear stresses, usually will result in a poor arrangement because suitable handholes cannot be cut without interfering with the structural characteristics. The ideal design at the engine section will provide a cowling which can be removed completely, exposing the engine so that adjustments, repairs or renewals, can be made readily. In this connection it is well to remember that any part of the powerplant which is difficult of access will receive a minimum of attention, and this may lead to engine failures in the air. It is necessary also that the engine shall be designed to expose all parts requiring frequent inspection. These questions are receiving and will continue to receive very careful attention in engine design.

The use of metal in engine frames will always hold a prominent place. The design of steel tubular A-frames is common, and such designs are structurally efficient. Frames developed from steel plates pressed into channel-sections also are becoming more common. Many practical difficulties are involved in this latter class, which result often in approximate failure of the design. In developing the channel sections designers may provide too little flange-depth and area, with the result that the sections are very inefficient about one axis. These frames usually are built in limited quantities which do not permit the development of suitable dies for forming the shapes. Forming is largely accomplished by hammering rather than by pressing. Anything but extreme care in this work will result in deformations of the sections due to dents and other unevennesses. This will cause local losses in strength properties. Gages used are often much too light. Gages common in tubular sections cannot be applied to other sections without further thought, for it must be remembered that the tube is a more efficient structural section than other forms.

The selection of suitable gages brings up intimate problems of design. It need hardly be stated that the engine frame cannot be designed for static loads only, and that due consideration must be given to the running stresses of the engine that are transmitted to the engine frame. It is necessary that the designer appreciate what these stresses are; thrust, torque and forces set up by vibration. He must develop his truss to accommodate the magnitude and direction of application of the forces involved. Without some special means for absorbing reversing stresses or those that vary rapidly in their intensity, rigidity of mounting is essential. Some supports will be developed with sufficient local strength and rigidity, but the structure between the engine and the truss anchorage will not have a proportionate amount of strength. Such design is equally faulty with that which is of insufficient strength and rigidity throughout. For single-engine installations it seems advisable to develop a rigidly jointed structure comprising the engine-bed and fuselage to the rear support of the wing, including the

center-section trussing and perhaps the chassis truss also. The importance of suitably mounting the engine cannot be overestimated. Airplane designs which have excellent flying characteristics but are designed poorly in the engine section and thus cause frequent failures of structural parts, will suffer an unnecessary loss in reputation because of these faults, which are largely due to a lack of appreciation of design requirements.

If an engine is to operate continuously at maximum power, it must have adequate cooling and be provided with an efficient means for temperature control. The cooling requirements of air and water-cooled types of engine are probably wholly different, but our experience with stationary air-cooled engines has been so meager to date that we are hardly qualified to write exact specifications for them. However, through winter flying experience had in different countries, particularly in Russia, where very low ground-temperatures exist, it has been found that air-cooled engines are the only type that can be operated successfully under such conditions. Rotary engines were used. The principal difficulties centered around the ability to start the water-cooled type before the water froze, even though it was supplied to the engine at a boiling temperature. Past experience with rotary engines has indicated with equal clearness that the air-cooled engine is heavily handicapped when operating under conditions of very high ground-temperatures, because of its inability to deliver maximum power required in taking-off. The need of the air-cooled engine for temperature control is an open question. There is nothing peculiar about this type to lead one to believe that it will not require such control. Radial engines have been flown by some foreign builders, largely without the provision of shutters. However, we do not know whether their engines operate efficiently at any and all temperatures, and so cannot judge whether this operation without temperature control is really successful.

A given design of water-cooled engine will require a definite capacity of cooling element when operating under the conditions prescribed by the performance of an airplane. The most important requirement in the design of the cooling system is the provision of the required size of radiator. Any effort to reduce the airplane drag by cutting the radiator size below the required capacity is decidedly short-sighted. Since the airplane designer will control largely the choice of position for the radiator, he must appreciate fully the advantages to be gained from a proper selection of core, from the standpoint of both water-flow and cooling efficiency. A detailed study of cooling systems is much too broad a subject to cover in this discussion. It is well to bring out, however, that a suitable choice of radiator type and position can result in the saving of about 35 per cent in the weight of this unit and about the same reduction in its drag.

The oil system for a given installation can be very simple. Generally speaking, it is unnecessary to include a special radiator for cooling the oil. Cooling can be effected more economically and the system can be simplified and lightened by exposing the oil-tank to a free flow of air. The design of the tank should insure that the oil is discharged against this exposed surface, and that the inlet and outlet pipes are arranged with respect to one another so as to insure complete circulation in the tank.

Nothing is more essential in the design of a fuel system than that it shall be direct and positive. Simplicity is required not only in the general arrangement of the system but in the means employed to induce flow through it. Because of the hazards of military flying, we find specifications written requiring the inclusion of two or more distinct systems. Such design is however not applicable to the needs of commercial flying. Both services will benefit by the use of absolutely reliable equipment for the fuel system, and the problem of supplying suitable accessories is now a serious one before the industry. A fuel-pump can and should be an integral part of the engine. Because of the extreme variation in the designs of fuel systems, this pump must have wide limits of ability. These limits will be well defined by the engine builder, and the system must be built to them. In conjunction with the pump there will be need for valves, strainers and the like, the design or type of which should be specified by the engine builder. Engine design will permit as much of this apparatus to be integral parts of the engine as is practical. All piping that can be made a permanent part of the engine should be installed so that only direct connections from the supply to the pump, valves or strainers will be required.

The proper functioning of the several systems is dependent on two things; first, a simple and correctly conceived design and, second, an ability to give these systems the mechanical attention they require. This latter ability is dependent on the skill of the designer to arrange his systems so that their essential parts are open to inspection, repair or renewal. Designers' attention is too often devoted to the problems of close assembly meant to conserve space, and not to the maintenance problems which will develop as soon as the airplane is in service. The value of extreme efforts to secure short-coupled airplane designs has already been questioned. The advantage secured by an airplane having features permitting easy maintenance is unquestionable. It must always be remembered that the value of a design is not determined as much by its ability to develop a performance, as by its ability to continuously and consistently give that performance; this latter condition can exist only when routine mechanical attention can be given the powerplant as a whole, with a minimum amount of effort and delay.

Summary

It is desired to emphasize the following points by summarizing them. We have not reached the limiting size for any type of engine as regards the maximum power available. No increase in engine performance can be expected unless new materials of construction, new fuels or new cycles of operation are made available. Continued development will refine the practices of the art and result in bettering the life of the engine and the service it renders, rather than its performance. Therefore, increased airplane performance must be secured mainly by improvement in airplane design. Great advance seems to be possible in this direction. One reason for the tremendous powerplants available for airplanes has been the effort to secure performance by brute strength. Absurdities can soon be reached if this trend of development continues. It is certainly worthwhile to consider what can be done with a reasonable-sized powerunit, by altering the design of the airplane. As shown in the last Pulitzer Trophy Race, excessive horsepower is not necessary to secure high speeds. The next few years should see a reduction in the power demanded of pursuit engines.

It is high time that attention be given to a most important problem, the fuel mileage obtainable from a given airplane. It is unquestionably true that the average person could not afford to operate some airplanes, even though he might be able to purchase one, because of the

poor mileage secured from a gallon of fuel. This is an essential consideration for commercial work, due to its effect on the profits of an operating company, and should be given study by the military authorities also, on account of the effect it has upon the quantities of fuel required in case of hostilities. This is, to our minds, the best and most obvious reason why the power requirements for aeronautical powerplants should be reduced rather than increased. It is not commercially possible to build so many power units within a given range. Airplane designers must be satisfied with fewer units, if we are to commercialize the business.

It is believed that the effect of engine dimensions on maneuverability is largely overestimated. The reason that popular comment is so often to the effect that the engine is the most important factor is the fact that there is no ready means by which the aerodynamic qualities of the airplane affecting maneuverability can be thoroughly analyzed and visualized. For similar horsepowers engine type rather than overall size will have the greatest effect on the parasite drag of the fuselage group. The efficiency of the cooling-element design for water-cooled engines is considerably better than that for air-cooled engines. Between practical limits, the effects on the performance of the airplane of variations in the values of engine weight per horsepower, of cooling efficiency, of fuel economy and of altitude performance of the engine are very pronounced. The demands of super-performance in military designs and greatest operating efficiency in commercial designs will require the development of engine types which are most favorable in these respects. The relative importance of the factors involved is governed by the particular service for which the airplane is designed.

Problems of powerplant installation are centered about the need of a close cooperation between the builders of airplanes and engines. The requirements of each system of installation can be met only by acquiring a correct knowledge of what these requirements are and satisfying them fully. A study of the engine mounting in complete detail, developing the truss system to accommodate all of the engine forces involved, both static and dynamic, is a most important requirement in insuring a successful installation. Simplicity and practicality of design and the suitability of the accessory equipment used are most essential in the development of the several powerplant systems. The development of the complete powerplant installation must be made with a view to permitting the greatest possible degree of service accessibility. Only in such a way can the proper mechanical attention be assured for the powerplant. The fact must not be lost sight of that the industry is in a formative period and for this reason we must expect to spend a tremendous amount of time and money in research. We cannot standardize without the necessary knowledge, and we cannot obtain that knowledge without research.

LOOKING INTO A GASOLINE-ENGINE CYLINDER

THE first tool used by the investigator is usually the eye. He sees a thing happen and the first step in his investigation is likely to be the obtaining of a telescope, microscope or something to enable him to see more clearly. In the development of the internal-combustion engine, however, visual observation of the combustion has played a minor role. To be sure, early investigators did provide apparatus that permitted them to look into the cylinder while the engine was operating, but little has been done in this direction with engines operating at the high speeds of the present-day automobile types. Interest has centered in what an engine could do rather than how it did it. As a result, measurements of brake horsepower and fuel consumption have been deemed of first importance.

At the present time, the cry for fuel conservation has reawakened interest in the nature of the combustion in the cylinder. Glass induction-systems have disclosed how satisfactorily, more often how unsatisfactorily, the fuel has been prepared for combustion. Analyses of the exhaust gases have shown how completely, more often how incompletely, the charge has been burned. Admitting that it is important to study the preparation of the fuel for combustion and the results which are evident after it has taken place, an actual study of conditions during combustion should surely be worthwhile. This has been accomplished in connection with the study of combustion in the one-cylinder Liberty engine at the Bureau of Standards. In conducting this work use is made of a spark-plug shell which has been adapted to receive a circular section of glass intended to serve as a window in the cylinder. This assembly can be used in place of either spark-plug in the ordinary aviation cylinder, but in this instance an additional boss has been welded to the cylinder in order that conditions might be observed with both plugs firing. Its use was satisfactory in that changes in flame color due to changes in air-fuel ratio were easily discovered. Since the entire combustion stroke is completed in 1/16 sec. at an engine speed of 1800 r.p.m., it is possible to see only the predominant color of the cycle by this means.

To make it possible to observe the combustion in its various stages, another device, a stroboscopic disk, was added. Its purpose is to permit the combustion to be observed during only a small portion of the stroke. Since there is one power-stroke for every two revolutions of the crankshaft, this disk is driven at one-half of the crankshaft speed. The flame is observed through a slot in the disk, the length of the slot governing the length of the portion of the stroke studied. Provision is made for altering the angular relation of this slot to the crankshaft so that any interval of the cycle and hence any stage of combustion can be studied.

Observations made possible by this apparatus are not likely to replace any of the more usual measurements. They may, however, prove, and in fact have proved of considerable value in research work of the nature described by permitting observations of the duration of luminous flame during the power-stroke, the characteristic differences in color and brightness at different phases of combustion and their variation with changes in ignition timing, mixture ratio, compression pressure, etc. One observes, for instance, excessively bright flashes of flame of extremely short duration accompanying the phenomenon known as fuel knock or detonation. The cause for this phenomenon is, of course, not revealed by usual observation.—Bureau of Standards News Bulletin No. 49.

Standardization as Brought About by the Society of Automotive Engineers[1]

By B. B. Bachman[2]

THE development of commerce from the dawn of history has been accompanied by the creation of standards, using the term in its broadest sense. The first were standards of measure to permit of the interchange of commodities, followed by standards of money as the use of money and credit displaced bartering. The growth of manufacture on a large scale, as distinguished from the making of goods to order, has brought about standards of sizes, as in clothing. The development of machinery to displace manual labor has brought about the adoption of standards for tools and equipment. It will thus be seen that unconsciously perhaps the crystallization of practice by continuous usage has been the means of creating standards which are used every day with so little thought as to their being or on what they were founded, that only the complete removal of them would give us an adequate conception of the absolute indispensability of these standards in our lives.

Time and again it has been pointed out that the automotive industry in the United States has reached its giant stature in so short a time as to be almost miraculous. The reason for this is manifold. Transportation has ever been one of the foremost needs of man, coupled with means of communication; any improvement in speed, dependability and comfort in transportation has met with instant acceptance and support by the public. The automobile, passenger car or truck, is primarily a means of transportation. Many have been the assertions by economists and others that the passenger car is fundamentally a luxury. Such assertions are, to my mind, refuted by the phenomenal growth of the usage of the automobile. The invention and development of the internal-combustion engine has made possible the creation of a light, compact prime-mover, which permitted the realization of the possibility of a transportation means for the individual over the public highways in the form of the passenger car and motorcycle, for merchandise by the truck, and later has satisfied the longing of man for the means of lifting himself above the earth and moving about without let or hindrance from the physical obstruction of mountain, stream and forest, and even the oceans which separate continents.

This marvel of mechanical science did not belong to us of America exclusively, however, and in many ways we were handicapped in the race to develop the automobile successfully from a curiosity to a point where it could be produced in quantities at prices which would put it within the reach of any but the plutocrat.

The successful solution of this problem lies in the art of quantity production of duplicate parts involving the use of machinery instead of men, and the interchangeability of parts and assemblies which facilitates and makes economical not only the manufacture of the article, but also its repair and maintenance. Carrying this thought to its logical conclusion, we arrive at the need for standards; and it is my opinion that the recognition of this need and the development and adoption of standards has been one of the most important reasons for the growth of the automotive industry. The industry was born coincidently with the beginning of the development of new methods of measurement, new machines and production methods. The men who started it were imbued with the pioneer spirit and would try once anything that looked promising, else they would never have started at all. In the beginning there was a great deal of rivalry and secrecy. Development was rapid and novelties introduced by one were copied or improved upon by others.

Beginning of Automotive Standardization

The old Association of Licensed Automobile Manufacturers was formed early in the history of the industry, but produced no real cooperation as realized today. Associated with this organization was the so-called Mechanical Branch, consisting of engineering and production representatives of the members of the Association, and while some progress was made in exchanging information among and creating standards for these members, such as the 7/8-in.-18 spark-plug thread and a series of bolt and nut sizes, the suspicion of the other fellow and his motives prevented a realization of all the benefits of cooperative endeavor.

Although this means of interchange of ideas existed, there were men whose vision reached out to a broader type of organization and a few of them got together and effected an informal organization in 1904 followed by the adoption of a constitution and the election of officers. These were A. L. Riker, president; Henry Ford, first vice-president; John Wilkinson, second vice-president, and E. T. Birdsall, secretary-treasurer. On the board of managers were H. M. Swetland, A. H. Whiting, L. T. Gibbs, H. P. Maxim, H. W. Alden and H. Vanderbeek.

From this beginning of the Society of Automobile Engineers, there was a gradual growth until in 1910 after the election of Howard Coffin as president, Coker F. Clarkson was secured as secretary and general manager and the beginning of a permanent office force effected. At the summer meeting in 1910, held at the Hotel Tuller, Detroit, the late lamented Henry Souther presented specifications for materials which were a revision of specifications which had been issued annually by the Association of Licensed Automobile Manufacturers. Later in this session discussion on the value of and necessity for standards culminated in the adoption of the policy looking toward the establishment of a Standards Committee, and this was effected later with Henry Souther as chairman.

Naturally the inauguration of such a policy was not a matter of unanimous consent. There were widely varying opinions, some of which have not been wholly harmonized today, and I feel I can do no better in presenting this phase of the subject than to quote from papers presented by Messrs. Souther and Clarkson.

[1] Presented at a meeting of the Automobile Accessories Business Association of Philadelphia, May 20, 1921.

[2] M.S.A.E.—Chairman of the Standards Committee, Society of Automotive Engineers, and chief engineer, Autocar Co., Ardmore, Pa.

Mr. Souther in 1912 presented his views as follows:

It is safe to say that all approve of some standardizing work and freely admit that some standards are possible. There is the class which believes that a standard, in order to be worth anything at all, must not be adopted or recommended until everything is known about the subject. In contrast, there is the class which seems to believe almost anything can be standardized and which would go to very great extremes in the matter. There is apparently a sharp division of opinion between these two groups. One believes that standardization should begin early in the history of an industry. The other believes that no standard is possible in an industry until such industry is so old that the probable changes in the proposed standard are few and far between. These views are diametrically opposite and require consideration.

Another view of a standard apparently taken by some contemplates the fact that a standard once adopted by the Society must be used by all its members or else they shall forfeit their rights to membership, or citizenship or something else not expressed. This is certainly not the case. A standard, in order to be accepted, must have so much merit that engineers or producers or executives will see good reason for its acceptance. Good engineering design may be the reason it is worthy of universal adoption; easy manufacture may be another reason why it should be adopted and become standard practice; and low cost is a reason why the business organization would say to the others involved that it must be adopted. As a matter of fact all three of these qualities will usually be found in any good standard. The adoption of a standard is not compulsory; it is voluntary.

There are many more details of an automobile that should not be standardized. To draw the line in a practical way between that which is fit material for standardization and that which is not, is the work of your Standards Committee and of the whole Society. Your committee will solve this difficulty satisfactorily, providing the members of the Society will come forward freely and frankly with whatever knowledge they may have on the subjects under discussion. A standard should certainly represent the collective knowledge of all concerned, rather than the narrow viewpoint of some one man or small group of men. The manner in which the Society of Automobile Engineers is attempting to arrive at proper standards is to get together all those interested in any way and promote such discussion as will result in some decision. Various interests must be involved. There are always at least two, the producer and the consumer. There may be several; for example, the producer of the raw material, the manufacturer who shapes it and finally, the consumer of the finished article. There is usually the sales interest between the other interests.

Mr. Clarkson's views along similar lines were presented in 1920 before the National Gas Engine Association, and are as follows:

One troublesome misconception of the standards we are discussing is that they are mandatory. Another is that they are manifestos of finality like standards of weight or measure. An automotive engineering standard is a thing that is considered, by men well qualified to judge, good or best for the great bulk of the manufacture in our field, to facilitate quantity production in the way I have indicated. The Society of Automotive Engineers has no way of enforcing the use of its standards except insofar as their merit is weighty. This is as it should be, and for a like reason the S. A. E. standards work has been successful. It has been demonstrated over a period of years that most of the standards can be reduced to practice by the great majority of manufacturers with marked benefit to themselves, as well as their customers; in fact, in all cases where the production is not really inherently special, or on account of large substantially identical previous production not incorporating the currently desirable standards. The latter condition is almost inevitably a matter of the relative importance of the past and the future to the manufacturer.

The Society of Automotive Engineers is not commercial in the sense that it can enforce its standards in an arbitrary way. It is commercial in the sense that its standards are of commercial value. The Society can conduct its activities on a somewhat broader and less partisan basis than a commercial organization. A commercial organization of manufacturers, proceeding as such, without giving effect to engineering questions as such, cannot, on account of competitive sales reasons, get as good results in the formulation of engineering standards as an organization like the Society can. In more than one instance that organization has established standards that have gone into general practice, after the representatives of the manufacturers directly concerned working together, or failing to work together, had been unable to establish them.

Standards should, of course, be canceled or revised when necessary. They should not, obviously, be promulgated originally unless there is sufficient evidence to assure their holding good for a properly long period of time. But the whole system should be conducted flexibly and not inflexibly.

Work of the Standards Committee

I do not feel that I need elaborate further on this, except to state that these views are a clear expression of the policy under which the Standards Committee has conducted its work. After a number of years of successful work, during which the growth of the Society was phenomenal, other organizations, recognizing the value of the Society in general, and the standards work in particular, became interested, with the result that in 1917 the Society of Automobile Engineers became the Society of Automotive Engineers and embraced, for purposes of fostering standards, the problems of the stationary internal-combustion engine, the marine and tractor interests, and aviation. From a small beginning with many problems of policy to solve, the Standards Committee has grown until for the current year there are 29 Divisions of the Committee, with a total membership of about 300, and a complete force under the general manager of the Society who devote their whole time to the work. Every endeavor has been made to conduct the work along the most representative and democratic lines, and furthermore, mere majority action is not allowed to carry through a proposal which a competent minority or even individuals oppose with well-founded objections. The results of the work have been highly satisfactory in both quantity and quality. There are at present listed in the S. A. E. HANDBOOK 215 separate Standards or Recommended Practices, divided into 10 divisions, covering Powerplants, Electrical Equipment, Parts and Fittings, Materials, Transmissions, Axles and Wheels, Tires and Rims, Frames and Springs, Controls and General Information.

As to the value of the work, much can be said, but unfortunately conversation has never established anything. The proof of the pudding is in the eating; and to my mind the best evidence of the value of S. A. E. Standards is shown by the fact that, in spite of great criticism and even sharp antagonism on the part of many,

(*Concluded on page* 38)

Eliminating Crankcase Dilution by Manifold Development

By G. P. Dorris[1]

Semi-Annual Meeting Paper *Illustrated with* Photographs

About 1912 the gasoline began to carry heavy ends in such amount that the application of a hot-water jacket around carbureters and manifolds improved the operation of the internal-combustion engine. Then the heavy ends increased up to 1915 and 1916, when a further revision became necessary to provide still higher temperatures than were possible with the limit of 212 deg. supplied by the hot-water jacket. Exhaust-heated manifolds and hot-spots helped to maintain the same degree of efficient operation with the then increased heavy ends of fuel. About 1917 the heavy ends of the fuel sold as gasoline required such an amount of heat to vaporize them that the expression "crankcase dilution" appeared. Now the heavy ends of the gasoline have increased to a maximum boiling point of 446 deg. fahr. This has made it necessary to go still further in the direction of the heat application to get satisfactory results.

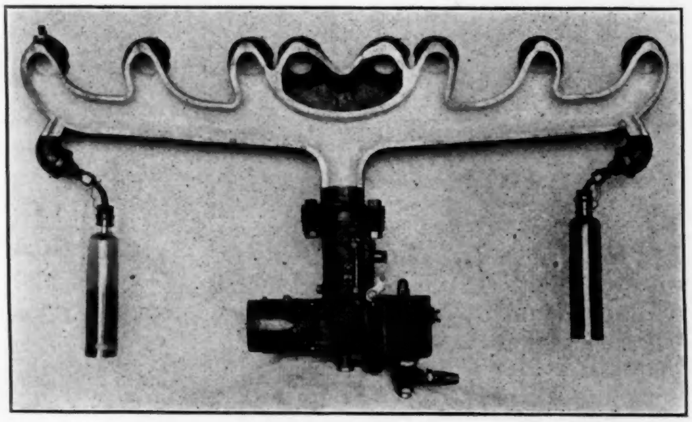

Fig. 1—An Admission Pipe of a Six-Cylinder Engine in Which the Two Exhaust Ports at the Center Form a Hot-Spot Immediately Above the Carbureter Inlet

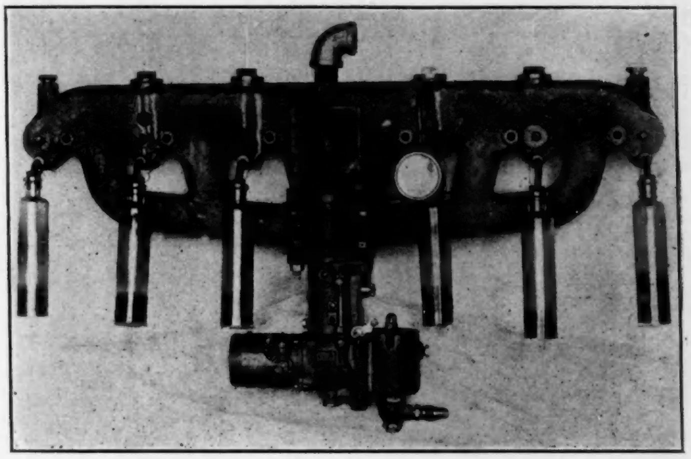

Fig. 2—An Improved Type of Manifold in Which the Admission Pipe Is Cast Integrally over the Exhaust Pipe thus Securing Better Vaporizing of the Heavy Ends

Relative Heat-Absorption of Air and Fuel

In the development of the internal-combustion engine it has become evident that the application of heat to the fuel through hot air is only a mild method of heat application and very detrimental to the volumetric efficiency and power output. It soon became recognized that the application of heat through the air was not a method of sufficient intensity to get desired results, and on further investigation it was seen that the direct application of heat to the fuel is a more efficient method as the relative heat-absorption capacity of air and fuel is approximately in proportion to their weights; hence the desirability of the elimination of hot-air jackets and the introduction of direct fuel-heating devices.

To the casual observer gasoline appears to be a definite fixed fuel not readily divided into several grades or degrees of combination of hydrogen and carbon. By subjecting this gasoline or fuel to a vacuum, it becomes evident that the vacuum readily separates the light ends from the heavy and gives a very satisfactory gas with a cold engine, which is very desirable in starting. Taking advantage of this fact, it is desirable to delay the delivery of the heavy ends of the gasoline until such time as the proper amount of heat can be applied to vaporize and pass

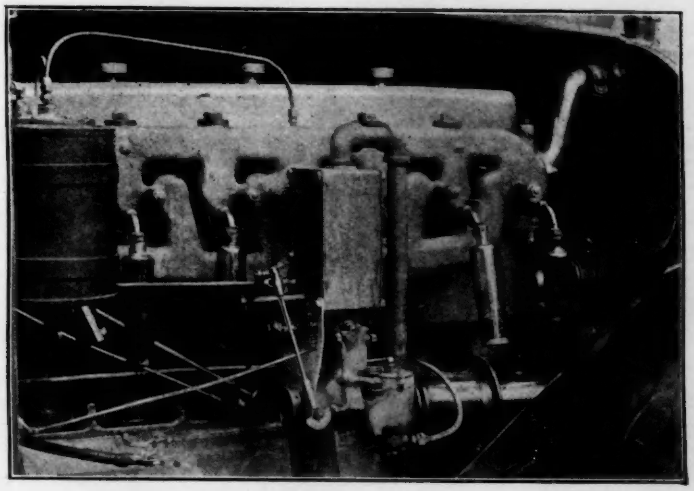

Fig. 3—Another Type of Admission Pipe Which Is Placed over the Exhaust Pipe and Separated from It by Air Spaces

[1] M.S.A.E.—President and chief engineer, Dorris Motor Car Co., St. Louis.

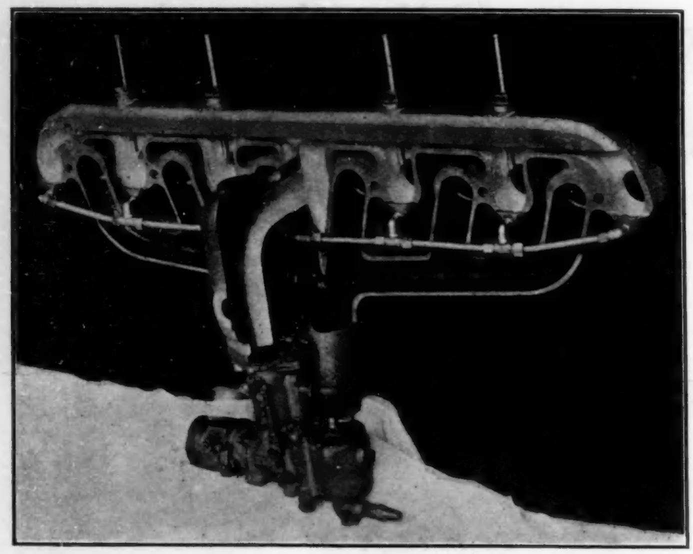

FIG. 4—ANOTHER VIEW OF THE ADMISSION PIPE SHOWN IN FIG. 3 WITH ONE-HALF REMOVED TO SHOW THE PIPE EMPLOYED TO RETURN THE HEAVY FUEL ENDS TO THE CENTER POCKET OR VAPORIZING STILL

them over to the cylinder in a readily mixed fuel for combustion.

TIME FACTOR REQUIRED FOR VAPORIZING

As the average motor-car engine requires approximately 1/25 sec., at a car speed of 30 m.p.h. to vaporize, compress and expand one charge, it is plain that the time required to vaporize the heavy ends is considerably longer than that available at this rate of operation; therefore it becomes very desirable to provide for delayed passage of these heavy ends so as to supply sufficient heat for their thorough vaporization, to use them as a vaporous fuel and to prevent any liquid fuel, or heavy ends, from entering the engine cylinders. The latter has caused disastrous results in crankcase oil dilution and premature wear in the engine bearings.

Fig. 1 represents an admission-pipe of a six-cylinder engine, the two center ports of which being exhaust ports form a hot-spot immediately above the carbureter inlet, so that all heavy ends are thrown against the hot wall between the exhaust and admission pipes. The light ends are passed on to the cylinder as a vaporous fuel and the heavy ends vaporized against this hot-spot. This pipe was satisfactory using the fuel which was obtainable from 1917 to 1919. For experiment, the ends of the manifold were tapped for drains, whereby an inspection could be made of the process of delivery of the fuel to the cylinders. In starting the engine at a temperature of 32 deg. fahr., 2 oz. of liquid fuel was collected in getting the engine warmed up to the operating temperature. The bottles would continue to fill slowly after the engine was warmed up. An inspection of the fuel in the bottles showed a 46-deg. Baumé gravity, while samples taken from the vacuum-tank supplying the carbureter showed 68 deg. Baumé gravity. This test proved conclusively that the fuel yielded freely to an evaporation of the light ends, leaving the heavy ends to trail along the bottom of the admission-pipe to the two end cylinders. From this it was obvious that a better utilization of fuel could be had under improved operating conditions.

Fig. 2 shows a revised manifold, wherein the admission-pipe is cast integrally over the exhaust-pipe, whereby considerable additional heat is applied and better vaporizing of the heavy ends accomplished. In making this pipe a pocket was provided between the two center exhausts, which is the lower pipe shown in the illustration, this pocket being an enlargement of the admission-pipe and set below the point at which the fuel enters the admission-pipe. This pocket collects considerable of the heavy ends on starting. The two center exhaust walls provided heat for vaporization after the engine was thoroughly heated up. Additional pockets were provided below the inlet ports, so that all heavy ends that might be projected over to the engine ports would be collected and their delivery delayed until such time as sufficient heat could be applied to vaporize them. To complete this experiment, bottles were connected to these ports and smaller amounts of the heavy ends were collected in proportion to the starting temperature and the use of the choker-valve in starting. Blind tubes were set into the admission-pipe down to the point of port-opening into the cylinder so that thermometers could be set into the pockets and the temperature of the gas at the point of admission to the cylinder noted. This temperature averaged 235 deg. fahr., which permitted the engine to operate in a very satisfactory manner, with very clean combustion and practically no carbon deposit, but, on account of the decreased volumetric efficiency, the engine lost power.

A third admission-pipe was made as shown in Fig. 3, it being placed over the exhaust-pipe and separated from it by air-spaces so that the admission ports were heated by being cast adjacent to the exhaust-ports, which secured a reduction in heat to the admission-pipe. Fig. 4 illustrates this admission-pipe with one-half removed and four of the thermometers in place. Six were used in the original experiment, but for production reasons the two end bosses have been eliminated. The thermometers show an average temperature, with the changed manifold, of 175 deg. fahr.

By utilizing a vacuum we vaporize a large portion of the light ends of the gasoline and do not heat the air or raise the admission-pipe temperature unduly.

Figs. 2, 3 and 4 show auxiliary heaters of different capacity bolted to the admission-pipe. These heaters are controlled by valves, more or less hot gases passing from the exhaust to them. Fig. 4 shows the admission-port pockets with the piping to return all heavy fuel ends collected to the central pocket or vaporizing still where they are stored until sufficient heat is accumulated to vaporize them.

With this type of manifold it has been possible to eliminate crankcase oil dilution completely and effect a reduction in carbonization. The lubrication efficiency has been improved. An improvement has been effected in acceleration and quick starting and the engine will run at a constant speed with a good torque within 1 min. after starting. A 4000-lb. car has covered 17 miles on a gallon of gasoline in a straight run at 30 m.p.h. In a 1-gal. test using kerosene as fuel a car equipped with this manifold made 17.1 miles. These results prove the efficiency of the equipment in preventing the passage of liquid fuel to the engine cylinders.

Vol. IX July, 1921 No. 1

Engineering Analysis Applied to Truck Selling

By N. J. Ocksreider[1]

Chicago Truck and Tractor Meeting Paper

When I read, write or think of the automotive industry, I segregate, almost invariably, the progress of this field into three 10-year periods, (*a*) that of experiment, (*b*) development and manufacture and (*c*) marketing. Most of the older people in the industry can recall readily the old one-cylinder type engine, built that way supposedly so that engine trouble could be more "readily located." The second 10-year period showed considerable development in 2, 4, 6, 8 and 12-cylinder engines. We are on the third 10-year period now marketing.

The old hit-and-miss idea of selling a prospective buyer of a motor truck anything he was willing to accept has disappeared. In its place the builder and the distributer have made great strides toward developing the selling of motor trucks on a thorough analysis basis. The builder must of necessity, to make his vehicles stand up, sell to the user the particular size truck that best fits the latter's requirements. In days gone by a truck salesman was too ready to sell a 2-ton truck to meet a particularly flexible requirement and as a result much trouble developed and brought about to a large extent lack of faith on the part of the user. When a truck is properly sold, it invariably gives far better results than a truck of the same size and make that has been improperly sold. The salesman must develop his method of selling along much broader lines. He must be able to observe and apply himself analytically. He must find the true problem as it exists and prescribe the proper remedy. Often an analysis of the customer's problems will show the salesman that the vehicle he sells cannot be used. He must then so inform the buyer and tell him what he should put in operation. The distributer and the builder must encourage this manner of selling through methods of training such as have been used by many of the larger manufacturers of specialties, including adding-machines, cash-registers and loose-leaf accounting systems. The motor-truck problem of today is not one of manufacture. There are questions of detail and refinement which the producer has yet to solve, but we know how to make remarkably efficient transportation machines. The great problem today, which invokes the honest consideration of the builder and the dealer as well as the user, is one of adaptability. What the user must decide is whether motor trucks or horses will best serve his needs, or whether a combination of these will produce the most desirable results.

The builder and the dealer must place themselves in the position of transportation experts, competent to give sound advice on questions arising in the field under discussion. Salesmen must be employed who are capable of analyzing a prospect's requirements and they must be encouraged to advise the prospect to act in his own interest, even at the risk of losing a sale. It is a great credit to the industry that there are cases on record where this attitude has been adopted and practised. It is obvious, of course, that the builder makes trucks to sell. It is precisely for that reason that he should see to it that his trucks are sold only where they can perform work most economically. This policy will inevitably develop "repeat" customers; any other will create enemies. The motor truck in its proper place is one of the greatest assets of modern life. Misplaced it is an economic waste.

Selling by analysis is not at all new. It has been in operation to my personal knowledge for more than 15 years; it is being used by nearly every large progressive industrial institution in America today. Some apply it more intelligently than others, but in a great majority of the cases the sales manager who is at all progressive realizes what the requirements of an analysis are. Charles M. Schwab very ably summed up the subject of selling by analysis in a single paragraph when he said:

> The super-salesman is a man true to the interests of his customer. His supreme purpose is to quicken the imagination of the customer and make him see the true virtue of the goods he is selling. The super-salesman foresees the needs of his customer and provides against those needs in full faith that the event will justify his foresight. He puts his ideals above his profits, in full confidence that profits will surely accrue to fine ideals intelligently executed.

There is nothing especially difficult about selling by analysis. It does not mean that a man must be a technical expert. It requires only a good supply of common sense, mental alertness, a fair amount of imagination and adequate capacity for work.

Market Analysis

In a Packard distributership where market anaylsis has been profitably used, a truck salesman had the following conversation with his manager:

> I am very sorry to be $700 behind in my drawing account. That is a lot of money to pay out and get nothing in return, but I think conditions will change.
>
> It is not the $700 that worries me, but the $6,000.
>
> You haven't paid me $6,000. I am only $700 behind in my drawing account.
>
> That's right. But what worries me most is the $6,000 in gross profit which we have lost due to the fact that you have not sold as many trucks in your territory as was anticipated. We assigned you part of the city with a thorough knowledge of its sales possibilities. We know how many vehicles are operated there now and in what lines of trade. We are further informed regarding the commercial activity in that section. We are not worrying about your drawing account, but about the business you are losing to competitors.

Do you know the potentialities of your territory for the sale of motor trucks in 1921?

How many of each size will you require during 1921, and in what months?

[1]Chief transportation engineer Packard Motor Car Co., Detroit.

What trades in your territory use the majority of motor trucks?

Do you know what share of the business is going to your competitors, and in what trades?

How do you determine quotas for salesmen? Can you prove that your territorial assignments are based on an equitable division of sales possibilities?

Do you know from which trades you can secure the greatest results with the least expenditure of selling energy?

Knowing the Facts About the Market

One of the most valuable assets in a business is the selling organization, and its effectiveness will vary in accordance with the available knowledge concerning the market. To profitably distribute any commodity it is necessary to analyze the market to determine the factors affecting it today and to anticipate future conditions. The whole trend of distribution is away from the old methods of trial and error and toward greater dependence upon scientifically determined facts.

Prosperity depends very largely upon accurate knowledge of the market. If there is a market in your territory for 5000 trucks in 1921, and you have a quota of 500, you must sell 1 in every 10. But if you erroneously anticipate a market for 10,000 and confine your efforts to selling only 1 in 20, the end of the year will probably find you with a warehouse full of unsold trucks. Distribution based on accurate knowledge of a market will reduce sales expense, because it eliminates useless expenditure of selling effort and directs it where it will produce the greatest results. Scientific distribution cuts down footwork and increases headwork. You can continue the old hit-or-miss methods, but are you sure that your competitors will not improve their methods?

Making a Market Analysis and Its Results

Two basic types of market analysis have been successfully applied to date by Packard distributers. The first type depends upon a study of motor-vehicle registration records according to geographical location, trade classification and make of truck. The second type is a personal canvass of a territory to determine all actual owners of motor trucks and to locate all potential prospects. This article will deal with the registration method. Secure an accurate motor-truck registration-service covering your territory. The information usually given is the license number, make, capacity and engine number of the vehicle, together with the name and address of the owner. Transfer the records to a card index. When all the cards are typed they should be filed alphabetically according to the owner's name. In this way all the vehicles belonging to each company will be automatically consolidated, and fleets of trucks will be represented as a whole. The next step is either to rearrange these cards according to the line of business or to make a duplicate set for that purpose. If thought necessary, a third set can be made and arranged according to the make of truck. There is no end to the information obtainable in this manner.

Among the advantages of market analysis may be mentioned:

(1) Knowing the trades active in each territory and estimating their probable requirements, your factory allotment can be accurately determined, thus preventing an excess of unpopular units and a paucity of units in demand
(2) Equitable assignments of territory to salesmen
(3) List of prospects for new trucks according to trades
(4) Prospects for second-hand trucks
(5) Comparative sales value of trades
(6) Active mailing-list according to trades
(7) Strength of competitors according to trades
(8) Where the manufacturer's trucks predominate, the information is especially valuable for sales use
(9) In trades where competitors' trucks predominate, steps can be taken to correct this situation

STANDARDIZATION AS BROUGHT ABOUT BY THE SOCIETY

(*Concluded from page* 34)

these standards and practices are so woven into the conduct of affairs in the industry that many who have not been in intimate association with their creation or who have not seen the lessening of troubles in drawing-room, factory, purchasing department and sales department because they were not familiar with conditions before the advent of the standards, do not realize that any other condition could exist. One of the greatest advantages of standardization is the fact that any shortcoming which develops and is corrected, or any improvement which is made is immediately reflected in the whole industry and is available to all, thus lifting the excellence and serviceability of the product of the industry as a whole. This is, of course, one reason why standardization is not applicable to proprietary or patented articles, and has its greatest value in items of form, dimension and material composition, where individual effort brings about inconsequential differences which are nevertheless differences and as such cost money.

One of the tendencies in modern business which is most notable is that of getting together, getting acquainted with the other fellow and his viewpoint. When this is done, we recognize that if we lock ourselves and our ideas up we will undoubtedly lock out much more than we lock in. I do not think I am stretching the point when I say that to the automotive industry belongs a large share of the credit for this change of view, and I hope you will pardon as harmless pride my further belief that the work of the Society of Automotive Engineers and its Standards Committee has been no small factor in showing the way in our industry.

As chairman of the Standards Committee, I can assure you that the policies which were laid down by the men who sponsored this work in its inception are being followed. We do not confine our consideration of suggestions and criticism to those received from members only, but hold our doors wide-open to any who have worthwhile matters to present.

You who have joined together in this organization and realize the benefits to be obtained from coordinated effort will, I am sure, appreciate the value of this work which the engineers of the industry have done.

Practice and Theory in Clutch Design

By HERBERT CHASE[1]

SEMI-ANNUAL MEETING PAPER *Illustrated with* DRAWINGS

THE objects of this paper are to (*a*) set down in convenient form for reference purposes particulars concerning American and British[2] practice in clutch design, (*b*) compare the advantages and disadvantages of various types of clutch and (*c*) give some notes on the theory of design without attempting a comprehensive treatment of the numerous factors involved in this theory. The descriptive portion deals almost entirely with clutches used on passenger cars and trucks, but some of the clutches described are applicable to other automotive uses. The notes on the theory of design apply in general to all automotive clutches. The clutches considered can be divided into the four general classes of cone, single-plate, multiple-disc and shoe or band types.

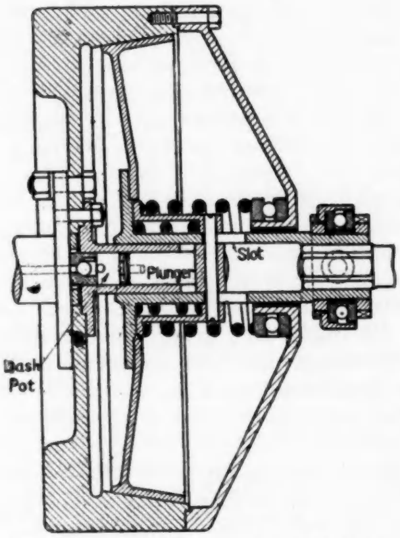

FIG. 1—CONE CLUTCH WITH DASHPOT

THE CONE TYPE

The cone type was used almost exclusively for many years for both passenger cars and trucks. It still is used very widely by Continental European and British builders, and has a considerable number of advocates in this country. It is simple in construction and can be made light enough to be brought quickly to rest when changing gear, at least in moderate capacities. In certain forms it is not expensive to manufacture and is fairly reliable; but, for a variety of reasons, it has steadily lost ground in this country and is used to-day by less than 8 per cent of car and 6 per cent of truck builders. In England over 60 per cent of car chassis and nearly 80 per cent of truck chassis still have cone clutches. It should be borne in mind that nearly all British builders make their own clutches, and that the average torque transmitted is much lower than it is in this country because British engines are smaller and, in the average case, of rather higher speed than American engines. Preference for the cone type in Europe is ascribed by some to simplicity and lower cost of production when the quantity is small. There are some who say that the cone type is ideal when properly constructed, and one British writer[3] contends that if the same ingenuity had been expended on the cone type as on the plate and disc types the first named would have proved so cheap, simple and satisfactory that other types would be considered expensive luxuries.

The chief failing of the cone type has perhaps been due to the fact that, as frequently constructed, the full pressure of the spring comes into action immediately upon engagement of the friction surface, thus causing the clutch to grab or pick up its load suddenly. To some extent at least this can be avoided by a variety of means. Among the simplest of these is the placing of subsidiary springs under the facing, so that the whole of the facing is not engaged simultaneously. A dashpot, similar to that shown in Fig. 1, also can be used, but this precludes sudden engagement which is sometimes necessary. Another expedient is to arrange two or more springs so that they come progressively into action on the cone during the period of engagement. Practice in respect to the angle of the cone varies widely, from 10 to 16 deg. in British types. The sharper the angle used, the lighter is the spring-pressure required to carry a given load with a given diameter and nature of surface. The clutch is likely to grab when the angle is made too fine, while a fairly wide angle tends to prevent grabbing but requires a heavy spring. The advantages of both are said to be realized in the construction shown in Fig. 2, in which a double cone is employed. The wide-angle surface engages first, while the narrow-angle face coming later into engagement furnishes sufficient friction to carry the full load without slip.

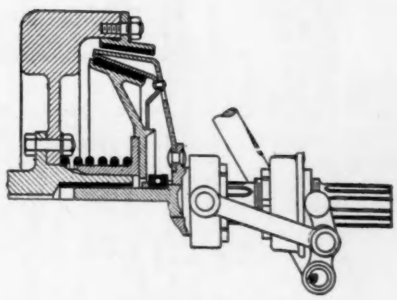

FIG. 2—DOUBLE CONE CLUTCH

Not many years ago all cone-clutch facings were made of leather, which often becomes hard and sometimes

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[2]The data and most of the particulars regarding British clutch practice were furnished by M. W. Bourdon, to whom acknowledgment is hereby made.

[3]See a paper entitled Developments in Transmission by Capt. S. Bramley-Moore which was presented before the Institution of Automobile Engineers and published in THE JOURNAL, April, 1921, p. 350.

glazed due to slipping on engagement. The clutch then either fails to carry the load or grabs until the facing is treated with oil. Too much oil lowers the friction co-

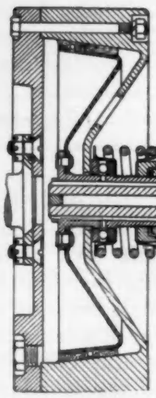


FIG. 3—THE 10-HP. HUMBER CLUTCH

efficient enough to bring about slipping and fullers' earth must then be applied to prevent slipping. Annoyance of this kind caused prejudice against the cone type and has favored the development of other types. Today, asbestos fabric is employed frequently for cone, as well as for other clutch facings. Cone clutches are sometimes allowed to run in oil, as in the case of the Humber clutch shown in Fig. 3. This keeps the facing material, which is leather in this case, relatively soft and pliable but of course decreases the friction coefficient and, other things being equal, requires the use of a stronger spring.

To facilitate gear-changing, it is desirable to decrease the moment of inertia of the driven member attached to the transmission; hence, its weight and diameter should be kept as low as possible. Some consider American practice superior to the British in this respect and say that gear-changing is easier on the American cars, but this may be due partly to other factors. To minimize

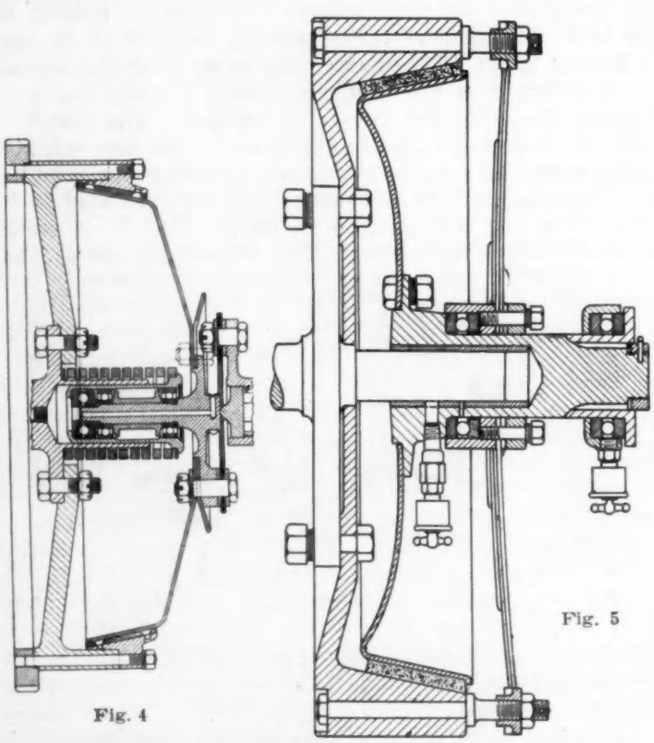


Fig. 4

Fig. 5

FIG. 4—THE CLEMENT-TALBOT CLUTCH
FIG. 5—THE THORNYCROFT CLUTCH

weight, either pressed steel or aluminum is employed for cone clutches. In some cases as in the Clement Talbot clutch shown in Fig. 4, the friction facing is attached to the female driving member instead of to the driven member, and the driven member is thus further lightened. This practice is said to have not been universally successful, due to difficulty in attaching the facing satisfactorily. In some cases facings are made and secured in segments, rather than in one piece, when they are carried by the driving member. The builder of the British Phoenix cars has followed the practice for many years of using an aluminum cone running in oil and bearing directly on the cast-iron flywheel. Phoenix cars are of small size, but the practice has apparently been successful, for it is followed even in the latest and largest 18-hp.-model of this maker.

Not all cones are of the conventional form, fitting directly into the flywheel. Some are of the so-called inverted type, such as the Humber, shown in Fig. 3, and are pushed forward toward the flywheel to disengage them. In this case a separate ring is required between the clutch and the flywheel proper. Nothing appears to be gained over the conventional type, unless it be in the compactness realized by placing the spring between the flywheel and the clutch. Nearly all springs used in clutch construction are either helical or volute types under compression. In many cases a single heavy spring centrally located is used, but two or more, equally spaced around the circumference near the periphery of the clutch are often employed. An exception is the Thornycroft clutch illustrated in Fig. 5, which employs two laminated leaf-springs, one on each side of the axis. These are fastened to the flywheel rim at each end by bolts, and make contact at the center with the ball thrust-bearing. The thrust of the engaging spring is usually taken by the flywheel-clutch unit; that is, the two or sometimes the clutch alone are self-contained, and no thrust is transmitted to external members except on disengagement. This is not true, however, in the case of the Daimler car, which has an exterior spring attached to the chassis frame applying pressure to the clutch and flywheel which must be taken up on the crankshaft. This construction is, of course, very much out-of-date.

So far as British practice is concerned, it appears that there is not much to choose between well-made cone and plate types. Their respective merits are so nearly the same that even experienced users accept either type without question. In other words, the type of clutch used is not a selling point in England. This can hardly be said to apply in this country, where perhaps the cone type has not been developed to the same extent as in British practice. The unsatisfactory performance of the cone type in some instances, and other factors, have led to the development of other types here, notably the multiple-disc type, which has advanced further and is more widely used in this country than in Great Britain or Continental Europe.

THE SINGLE-PLATE TYPE

The single-plate type of clutch is used widely in both this country and Europe. When well made and properly adjusted, it is smooth-acting. It is of simple construction and can be made with few parts; consequently it lends itself to economical production in quantity. As compared with the cone and the multiple-disc types, heavier spring-pressure is required to carry the same torque; hence, if the same linkage is used, the pedal-pressure necessary to disengage the clutch is greater,

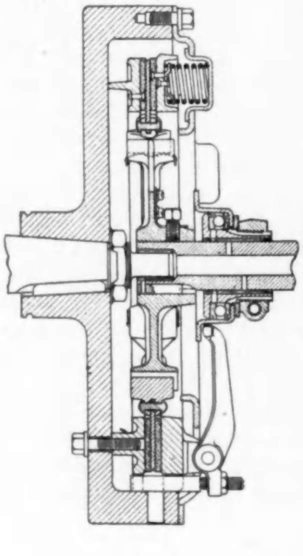


Fig 6

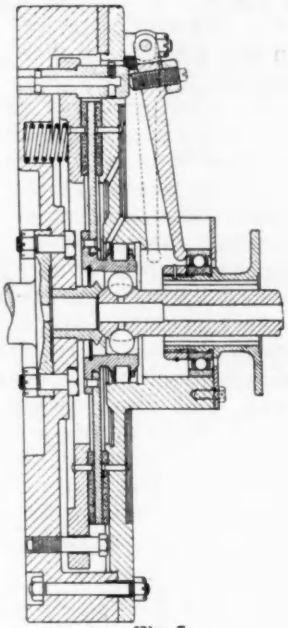


Fig. 7

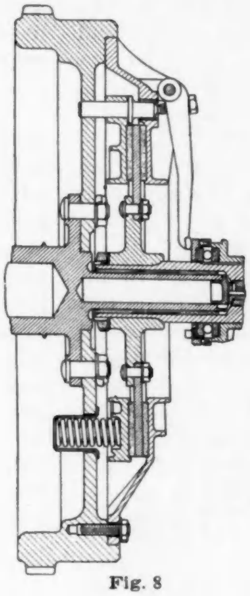


Fig. 8

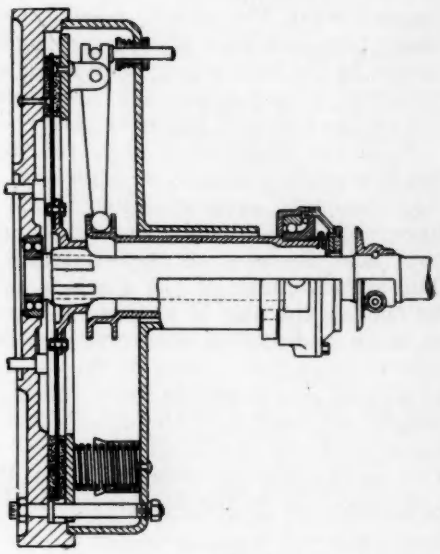


Fig. 9

Fig. 6—The Autocar Clutch
Fig. 7—The Dennis 2-Ton Clutch

Fig. 8—The 40 to 50-Hp. Napier Clutch
Fig. 9—The Bristol 4-Ton Clutch

unless a much larger diameter of plate is employed. For this reason multiplying levers are employed in the throw-out mechanism, and many different arrangements of levers are used in plate clutches of different makes. In some cases the levers are attached to and rotate with the driving member, while in others external leverage, aside from that on the pedal lever, is employed.

Figs. 6 to 13 show plate clutches using multiplying throw-out levers. Some of these are employed on trucks and some in passenger-car service. A somewhat different design of lever is employed in all of these eight clutches. In the case of the Autocar, Fig. 6, the Dennis, Fig. 7, and the Napier, Fig. 8, adjusting screws which can be backed off as the facings wear are provided on the levers. The Bristol, Fig. 9, the Halley, Fig. 10, the Mack, Fig. 11, and the Austin, Fig. 12, appear to have no provision for adjustment within the clutch itself. On the Arrol Johnston, Fig. 13, the spring-pressure can be varied by turning the threaded spring-caps, but the levers are non-adjustable. In all of these eight clutches three or more springs, equally spaced in a circle having a radius approximately equal to the mean radius of the friction disc or discs, are employed. These are so-called "direct-acting springs," since their pressure acts

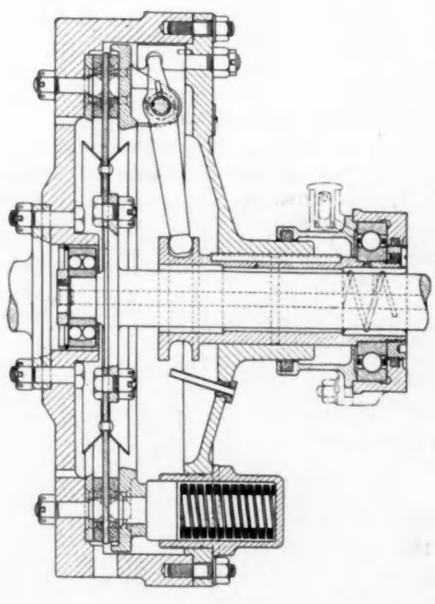


Fig. 10—The Halley 3½-Ton Clutch

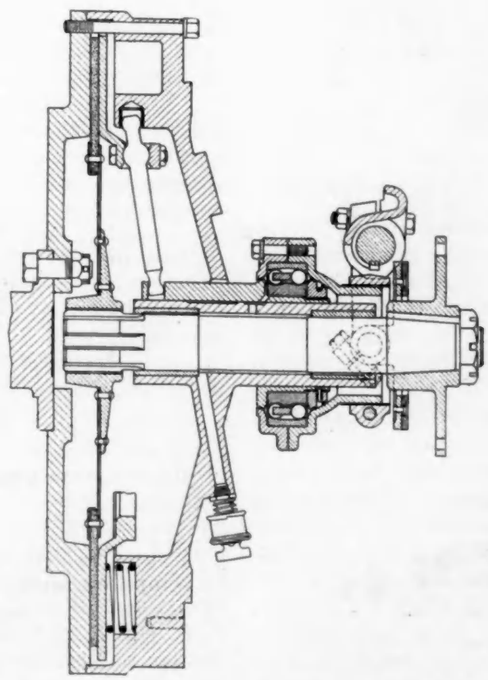


Fig. 11—The Mack Clutch

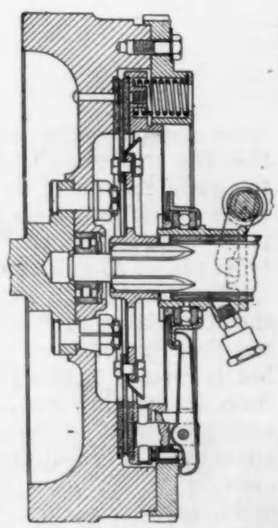


Fig. 12—The 20-Hp. Austin Clutch

directly upon the friction surfaces and not through levers, as in the types that will be described later. As the facings wear, the springs become somewhat longer and their pressure decreases correspondingly, but the increase in the length is usually small and, if the spring is properly proportioned, the decreased pressure is still within the capacity factor of safety allowed.

With the single-plate type, the unit pressure on the friction surface usually is high and, since the slip with its consequent abrasive action which takes place during engagement and disengagement occurs as a rule only on two faces, the wear on these surfaces is relatively rapid. This disadvantage of the single-plate type is minimized by making the disc of as large an area, and consequently as large in diameter, as is feasible; but, when the diameter is large, the inertia of the driven member becomes great and gear-changing more difficult. To minimize the weight and inertia of the driven disc, it usually is made fairly thin and the friction surfaces are, as a rule, carried on the driving members. The thin plate is noticeable especially on the Bristol clutch shown in Fig. 9. In useful life of the friction disc is much increased. A somewhat different type of single-plate clutch widely used in this country is shown in Figs. 14 to 16. This type employs a single spring placed co-axially with the clutch itself and arranged so that the pressure of the spring on the facings is multiplied by lever and toggle devices. The advocates of this type claim the advantage of uniform pressure at all points on the friction surface, which, they contend, is not secured when separate direct-acting springs are employed. This advantage is, however, dependent upon having the same size and fit on the various bearing parts involved, as well as on equal wear; hence, there is some question whether a more uniform pressure is obtained in reality than is secured by properly calibrated sets of springs acting directly on the friction faces. One disadvantage resulting from the use of levers to multiply spring-pressure is the fact that the motion of the long end of the lever due to wear on the facings is multiplied in the ratio of the lever arms. If this ratio be 5 to 1, wear of 1/16 in. on the facings means a 5/16-in. extension of the spring. In some designs this

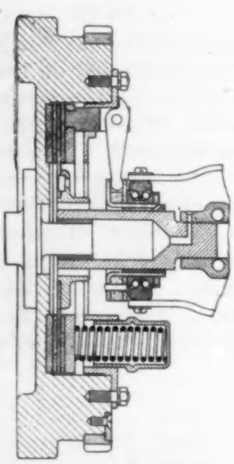


Fig. 13—The Arrol Johnston Clutch

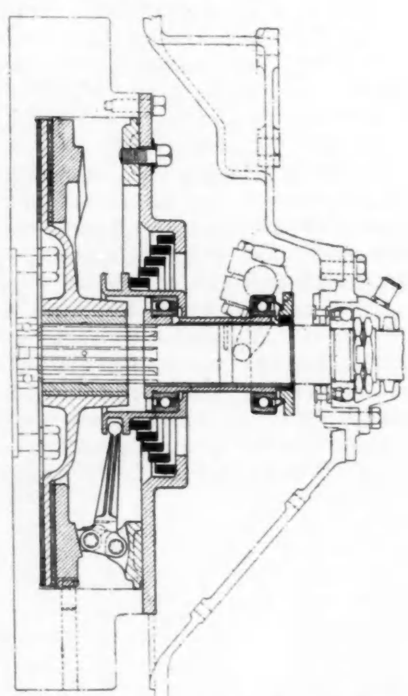


Fig. 14—The Borg & Beck Clutch

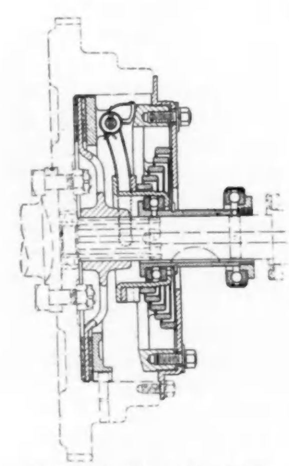


Fig. 15—The Hoosier Clutch

some cases, however, so thin a section has been employed that the heat caused by slipping has resulted in warping the plate enough to require its renewal. High temperatures are often reached when, as in traffic driving, slipping occurs frequently. Heat can radiate but slowly from plates so fully enclosed.

The Mack clutch illustrated in Fig. 11 is designed so that the driven friction disc only is clamped between the two driving-members. It is carried by a light steel plate, but is riveted to this plate at points nearer to the center than the bearing surface. This arrangement, no doubt, adds somewhat to the moment of inertia, but it has the advantage of requiring only a single disc of friction material which takes the wear on both sides. Furthermore, as wear occurs, the rivet-heads never bear on the metallic friction-surface as in most other constructions after considerable wear takes place. In this way the amount of motion will involve sufficient reduction of pressure to permit slipping and, since a motion of the throw-out sleeve of about ½ in. is usually allowed, it is necessary with this type to adjust with relative frequency. Hence, clutches of this type are, as a rule, provided with means for quick adjustment.

In the Borg & Beck clutch shown in Fig. 14 adjustment is made by turning the ring on which the toggle-levers are carried, thus causing the inner ends of the toggles to bear sooner on the helical surfaces provided on the thrust ring. The toggle-ring is then locked in place by two cap-screws passing through the cover-plate. In the Hoosier clutch, illustrated in Fig. 15 a similar adjustment is made by turning the threaded adjusting-ring, which is then locked by cap-screws in a manner similar to that used in the Borg & Beck clutch. The release sleeves used in these two clutches are attached to

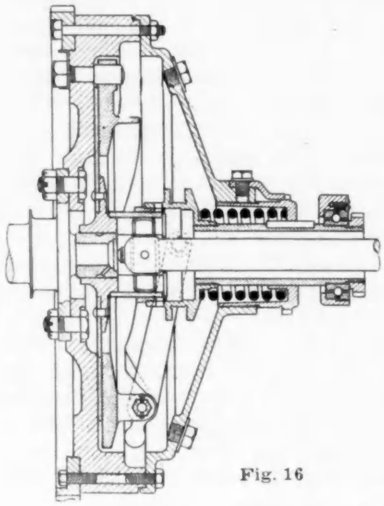


Fig. 16

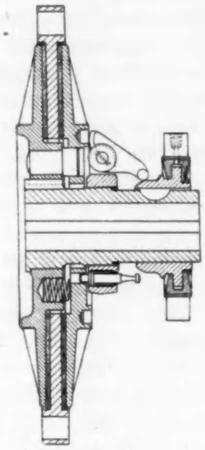


Fig. 17

FIG. 16—THE ROVER CLUTCH
FIG. 17—THE TWIN-DISC CLUTCH

and turn with the driven member, while the collars against which the toggle levers bear at their inner ends turn with the driving member. This necessitates the use of an extra bearing between the two, which is not required in most other constructions. The facings in these two clutches are allowed to float; that is, they are not attached to either the driving or the driven members. Since no rivets are required, the facings can be used without renewal until nearly worn through.

The earlier types of multiple-disc clutch ran in oil and used alternate discs of bronze and steel. With the exception of the Phoenix cars which, as stated above, use aluminum cone-clutches running in oil and bearing on the cast-iron flywheel, few other metal-to-metal clutches are used to-day. One of these is the Rover, shown in Fig. 16, which has a single phosphor-bronze plate running in oil between two cast-iron surfaces. Levers which multiply spring-pressure on the friction surface are employed as in the case of the last two clutches mentioned, but these are not of the toggle type in which a certain wedging action occurs. The use of oil, of course, reduces the coefficient of friction, and this necessitates greater pressure than would be required otherwise. Other disadvantages encountered when disc or plate-type clutches run in oil are cited under the head of multiple-disc clutches. In the case of the Rover clutch the torque transmitted is only that of an engine of 12-hp. rating; hence, it would seem to be possible in a clutch of the proportions shown to work with a large factor of safety, and this is no doubt the case, for the clutch is said to work very well in practice. The Rover clutch is designed for use with a separate transmission; hence, a universal-joint is employed at its center. The disengaging sleeve is keyed to the driving member and therefore rotates with it. Through this sleeve passes the shaft of the universal-joint which, of course, turns with the driven member; but, since this shaft does not touch the sleeve, no bearing is required between them and the thrust of the spring is taken against the casing direct. This makes the extra bearing used in the Borg & Beck and the Hoosier types unnecessary. Spring-pressure can be varied by turning the threaded sleeve, against which it bears, in or out of the conical housing. The bronze driven member apparently wears so little that no other adjustment is required.

A variation of what is essentially a single-disc type of clutch is shown in Fig. 17. This is known as the Twin-Disc clutch and has a pair of driven but only one driving plate, thus reversing the usual single-plate arrangement and materially increasing the weight and inertia of the driven member. No engaging springs are used in the clutch. It is of the locking-toggle type and consequently must be positively engaged and disengaged. For this reason it is not suited to or intended for use on automobiles or trucks, but can be used in some tractor applications. It is adjusted easily by disengaging the spring-retained pin projecting from the threaded toggle carrier-ring, and then turning this ring until the pin falls into another hole on the driven member.

THE MULTIPLE-DISC TYPE

The multiple-disc clutch is regarded by many American engineers as the best of all types for both passenger-car and truck service, as is evidenced by the fact that it is used on nearly all of the more expensive cars produced here. This type of clutch has apparently been developed to a higher degree here than in Europe, while there other types, as previously pointed out, have seen the greater development. Partly for this reason, the multiple-disc type is not used widely either in Great Britain or in Continental Europe, although some car-builders there use it with evident satisfaction. The multiple-disc type of clutch in most forms is more expensive to manufacture than other types, hence, its use has been more general in the higher-priced cars. It is used, however, in many low or moderate-priced cars, in the Ford and the Dodge cars for example, and is not inherently expensive to produce.

The earlier types of multiple-disc clutch were of the metal-to-metal variety and used either all hardened-steel discs or alternate discs of steel and bronze, or steel and copper. Some clutches of this type are still used, such as the Vauxhall, which is shown in Fig. 18. With this construction it is necessary for the clutch to run in oil. This permits a smooth engagement but the use of oil results in more or less trouble, for satisfactory operation of this variety of multiple-disc clutch depends upon having both the correct viscosity and the proper quantity of lubricant. Too much or too viscous lubricant will tend to prevent proper engagement and consequently cause

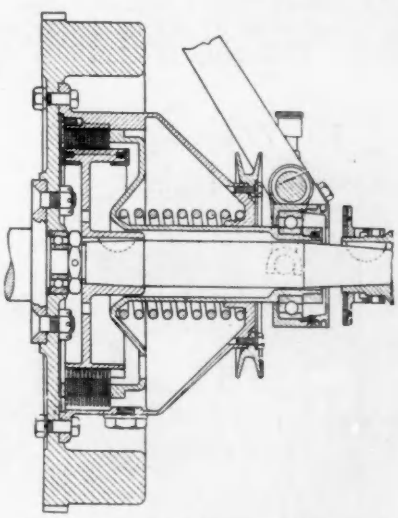


FIG. 18—THE VAUXHALL CLUTCH

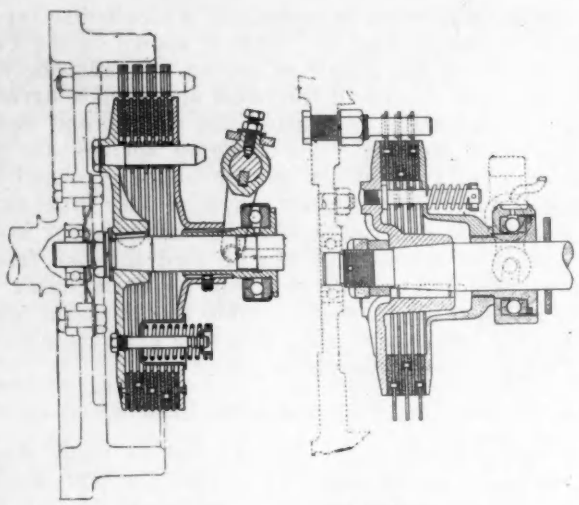


FIG. 19—THE HUPMOBILE CLUTCH
FIG. 20—THE DETLAFF CLUTCH

slipping, and will tend also to make adjoining discs adhere when disengagement is required, thus causing the clutch to drag and render gear-changing difficult. Lubricant which is of too low a viscosity or is present in too small a quantity is apt to result in grabbing or cutting of the plates. When oil is used the coefficient of friction is, of course, very much reduced and in consequence many more plates are necessary for a given pressure and diameter to carry a given torque than when dry discs are employed. In consequence, the clutch which runs in oil is, for the same capacity, usually more expensive and heavier than the dry multiple-disc type. For these reasons a great majority of the multiple-disc clutches used to-day have discs of steel, with alternate discs faced with asbestos composition; these discs run dry, and are frequently not enclosed. Clutches of this type prove highly satisfactory in service when well constructed and properly proportioned. They are smooth in engagement, require practically no attention throughout the life of the facings, which last, as a rule, from 20,000 to 50,000 or more miles of car operation, and then are replaced easily. This type of clutch is compact and,

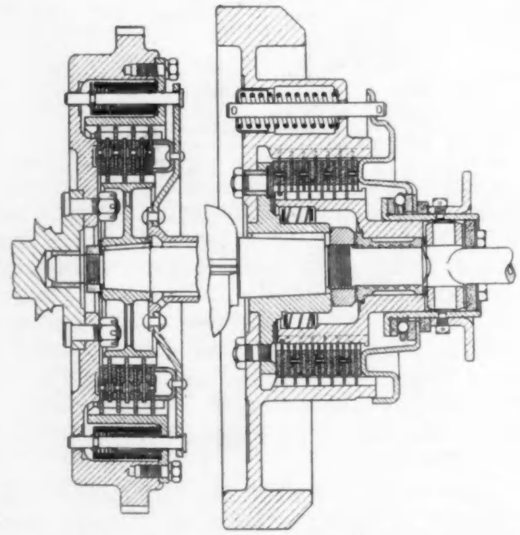


FIG. 21—THE CLUTCH USED ON THE PACKARD SINGLE-SIX CAR
FIG. 22—THE CLUTCH WHICH IS USED ON THE REO PASSENGER CAR

since the driven members are light and of small diameter, gear-changing is facilitated. While the number of parts used is greater than in other types, most of these are duplicate stampings, easily and cheaply made; hence, the cost of production need not be high and the clutch is readily made in a self-contained easily removable unit. The large friction surface obtainable in multiple-disc construction makes for long life, and the large number of surfaces permits the use of relatively light spring-pressures and consequent easy action. In some cases fairly heavy springs and a small number of discs are employed without apparent advantage, except perhaps in the matter of cost.

Two distinct classes of multiple-disc clutch are recognized in the American trade; the pin type, in which the drive or torque is taken through pins attached to the driving and driven members, and the gear-tooth or key type, in which gear-teeth, keys or their equivalent take the drive. The Hupmobile clutch shown in Fig. 19 and

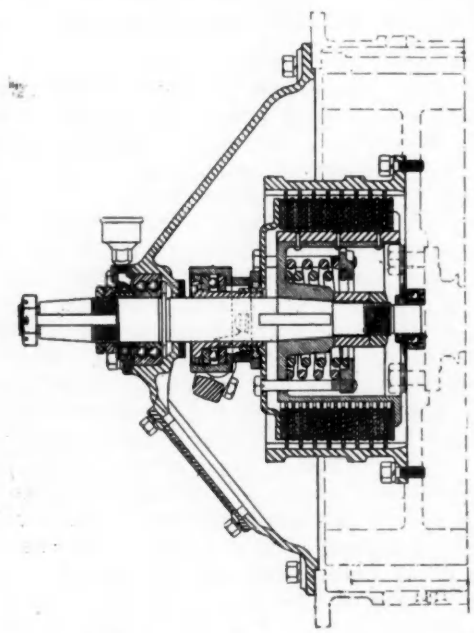


FIG. 23—THE G. M. C. TRUCK CLUTCH

the Detlaff clutch illustrated in Fig. 20 are examples of the pin type. The Detlaff company makes also a gear type that is not shown here. It will be noted that the driving discs engage with driving-pins, usually three in number, attached to the flywheel, while the driven discs engage with similar pins carried on the rear driven-member which is, in turn, keyed or splined to the driven shaft connected to the transmission. In the Detlaff clutch the driven pins also carry the engaging springs, but in the Hupmobile design the springs are on separate pins, making a more compact layout, but adding extra parts. The friction facings in one case are riveted to the driving-plates; in the other they are riveted to the driven plates, adding, it would seem, unnecessary weight to the driven unit. In most if not all cases the pin type of construction is less expensive and probably somewhat lighter than the gear or disc type, but it has one rather serious disadvantage as ordinarily constructed.; it is apt to become noisy when wear takes place, because the holes in the discs through which the driving and driven pins pass cannot be made to fit the

pins tightly, owing to the fact that the discs must be free to slide upon the pins and that the discs must be free to expand under heat and consequently must have sufficient clearance to allow this expansion. Since the hole is either somewhat elongated or of larger diameter than the pin, only line-contact between the two is possible without deformation or wear. Since the discs are relatively thin and the pressure due to torque is considerable, wear does take place and chattering frequently results when the clutch is disengaged.

Efforts to minimize wear are frequently made by providing larger bearing surface. This is done in the Detlaff design by punching the holes with a lip turned outward. In the Hupmobile clutch bushings are attached to the discs at the driving-pin holes, but none are provided at the driven pin holes where, of course, the torque and pressure are greater. However, so long as round pins are employed, only line-contact is theoretically possible. In some cases more than three pins are used, thus increasing the number and total area of contact surfaces. The gear or key-type clutch goes another step further in this direction by increasing the number and area of the contact surfaces still more. Examples of the gear type of construction are the Packard, shown in Fig. 21; the Reo, in Fig. 22; G.M.C., in Fig. 23; and the Hilliard, in Fig. 24. The Locomobile clutch shown in Fig. 25 is an example of the key type, while the Merchant & Evans clutch in Fig. 26 is a combination of gear and key construction. The Browne clutch shown in Fig. 27 uses a straight-side tooth on the driving disc, while the driven discs bear on substantial cast lugs. In clutches of these types an external driving-ring, which can be an integral part of the flywheel, and an internal driven drum generally are employed. As a rule, the ring has either internal teeth cut therein or keys usually secured to its internal surface by rivets. The driving-plates are formed to mesh with the ring teeth or keys, with sufficient clearance to slide freely in an axial direction. The drum has teeth or keys cut in or attached to its external surface, and these teeth, meshing with corresponding teeth or recesses in the driven plates, transmit the torque to the driven shaft to which the drum is splined or keyed.

The discs will bear equally on each tooth when well-

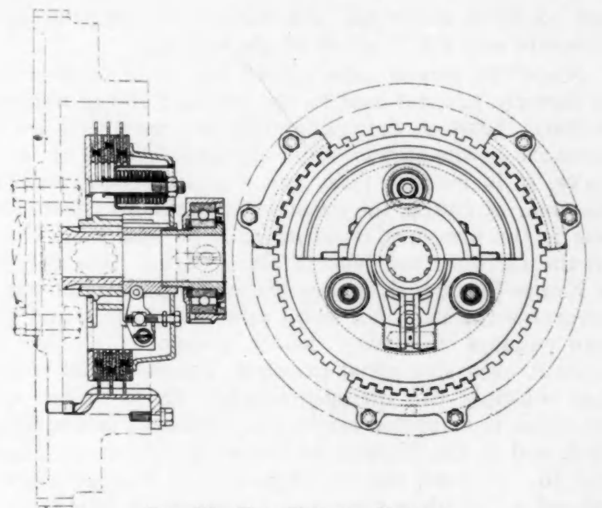


Fig. 27—The Browne Clutch

cut gear-teeth are employed, thus reducing the unit pressure and wear as compared with the pin type. In the key type each key will bear equally over the full area of the notched surface of the disc, when properly made. Since the number of bearing surfaces can be made large, the aggregate surface is considerable and the unit pressure is much decreased relatively. In the Browne clutch shown in Fig. 27 the key-type construction is carried a step further without using keys. The driving plates have straight-side teeth which mesh with similar teeth broached in three separate gear segments attached to a standard S. A. E. flywheel-rim, while driven plates have a large bearing surface on flat-faced lugs cast integrally with the driven spider. The segments are stampings easily and cheaply produced in quantity; their use makes it unnecessary to use an expensive ring-gear with internally-cut teeth. Both the gear and the key types are apt to be less noisy than the pin type, but in general they are more expensive and are therefore used, as a rule, only on the more expensive and higher-powered cars and trucks. They are better suited than the pin type to carrying a large number of plates and usually are able to transmit a higher torque. It will be noted that practice varies considerably in respect to the num-

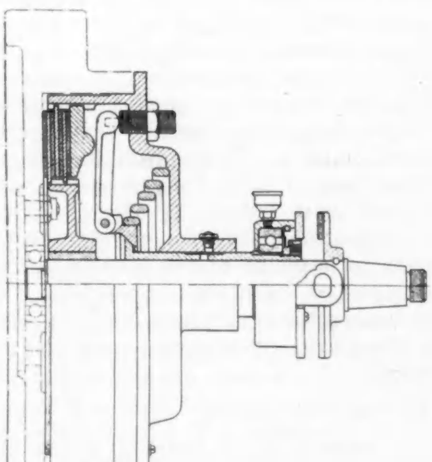


Fig. 24—The Hilliard Clutch

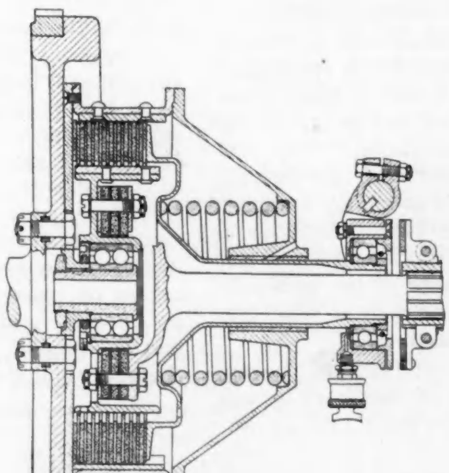


Fig. 25—The Locomobile Clutch

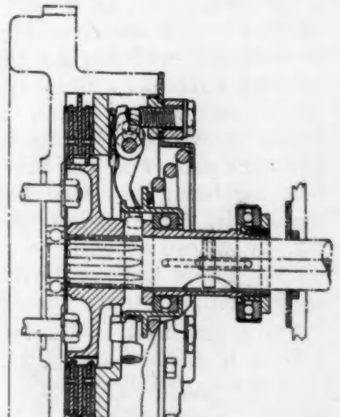


Fig. 26—The Merchant & Evans Clutch

ber of discs employed, the method of applying spring-pressure and the location of the springs.

Since the torque capacity of any plate or disc clutch is directly proportional to the product of the number of surfaces between driving and driven members, the total pressure on the surfaces and the mean radius of the surfaces, it is evident that, for a given torque and mean diameter of clutch, the greater the number of discs is, the less is the pressure required and the less wear there is on the facings; the reverse also is true. The Merchant & Evans Co., which makes the clutch shown in Fig. 26, advocates the use of a small number of discs and therefore employs relatively heavy pressure. To get this without excessive pedal pressure, resort is had to levers that multiply the spring-pressure. In so doing, a construction is employed similar to that used in the Borg & Beck and in the Hoosier plate-designs shown in Figs. 14 and 15. In fact, the clutch shown in Fig. 26 might be termed a two-plate type and is subject to the same disadvantages cited in connection with Figs. 15 and 16, since it requires an extra bearing and relative frequency of adjustment, but it has a larger wearing-surface than the two types with which it is compared.

The other multiple-disc clutches shown employ a large number of discs and, with the exception of the Hilliard clutch which in some cases runs in oil and uses multiplied spring-pressures, have direct-acting springs giving lower total pressures on the friction surface than other types. In the G. M. C. truck clutch shown in Fig. 23, two centrally located springs are employed. In this case the outer end of the clutch shaft is carried in a ball bearing supported by a bell-housing attached to the crankcase. This latter is conventional American practice where a separately mounted amidship transmission is used.

The Locomobile clutch shown in Fig. 25 is unconventional in that the double-row ball pilot-bearing is carried on a spigot extending from the flywheel, whereas this bearing is usually recessed within the flywheel, and in that it incorporates within the clutch a fabric universal-joint. From this joint the driven shaft extends through a tubular extension of the pressure plate, but does not touch the latter. No bell-housing is necessary in this case, although a separate amidship transmission is used. A single central spring is employed. Another rather unconventional construction which incorporates a universal-joint as a part of the clutch unit is that of the Reo clutch shown in Fig. 22, in which the crankshaft is extended well beyond the clutch where a plain pilot-bearing supports a block-type universal at the end of the shaft connecting clutch and transmission. The ball thrust throw-out bearing is placed between the driving pressure-plate and the driven throw-out collar, no other bearing being required between the two. Pressure is applied by three springs carried within the flywheel and equally spaced around the circumference of the pressure-plate. There are six driven plates and only a relatively light spring-pressure should be required.

The clutch used in the latest model of Packard passenger-car is shown in Fig. 21. This is of the more or less conventional design used in connection with unit power-plants. There are but three driven plates; hence, heavier spring-pressure is required than would be necessary with more plates and the same torque. This pressure is supplied by four springs carried between the driven drum and the flywheel. An external multiplying linkage is employed between the clutch and the pedal to decrease the pedal-pressure required to disengage the clutch. A plain pilot-bushing is used within the crankshaft flange. The Browne clutch shown in Fig. 27 is one of the most recent designs of multiple-disc clutch. While it deviates but little from conventional practice it incorporates several refinements in detail which indicate thorough appreciation of the problems involved both in use and in manufacture. For example, multiplying levers are used to decrease the throw-out pressure, which makes for light pedal-pressures without external multiplying linkage, but the springs are direct-acting and proportioned so as to require no adjustment during the life of the facings. The springs are contained within a pressed metal cup and carried on a thin metal tube which is turned over at its ends to prevent the spring from attaining its free length when the spring-bolt nut is removed. The length of the tube is such as to enable the nut to catch a few threads on the bolt before seating on the spring. This greatly facilitates assembling, especially when the clutch is dismantled in a repair-shop for the renewal of facings.

Adjustment is seldom required in the multiple-disc type of clutch with direct-acting springs, and then is made usually by turning up the nuts on the spring-bolts. In the Browne clutch another and very convenient adjustment is provided. The multiplying throw-out levers each carry a threaded bolt with two heads. The outer head projects through a slot in the cover-plate, and is adjusted to bear on the plate. The inner head bears on the inner surface of the cover-plate when the clutch is disengaged. As wear takes place in the friction discs the plate moves inward, leaving the bolt-heads projecting above the surface. When they are seen to project about 3/16 in., they are simply screwed in until they again bear on the plate and the clutch is then again in adjustment. Such adjustment is required only once or twice during the life of the facings. Allowing all the bolt-heads to bear on the plate assures uniform adjustment of the levers.

THE BAND-AND-SHOE TYPES

Since band-and-shoe types of clutch are little used in automobile and truck applications, no special effort has been made to collect designs of this character, but one of the shoe type known as the Pfeiffer clutch is shown in Fig. 28. This consists of two semi-circular shoes covered with asbestos fabric and arranged to engage the inner surface of the flywheel rim under influence of the spring and toggle mechanism shown. The action is somewhat similar to that of an internal expanding brake. The friction surface is fairly large and, since it is disposed at a rather large radius, the clutch should have high torque-capacity with the multiplying linkage shown, even though the spring used be light. The weight and inertia of the driven member are considerable, however, and therefore gear-changing will not be facilitated, while somewhat frequent adjustment of the toggles would doubtless be necessary on account of the large multiplication of pressure employed. Clutches somewhat similar to external-band brakes have also been used in automotive vehicles, but these are seldom seen to-day and are therefore not described.

DETAILS OF DESIGN

All clutches require some type of bearing which will take the thrust imposed by the throw-out mechanism in disengaging. Ball thrust-bearings are frequently employed. These are relatively cheap and give satisfaction when not required to carry radial as well as thrust loads, especially if the construction is such that they run only

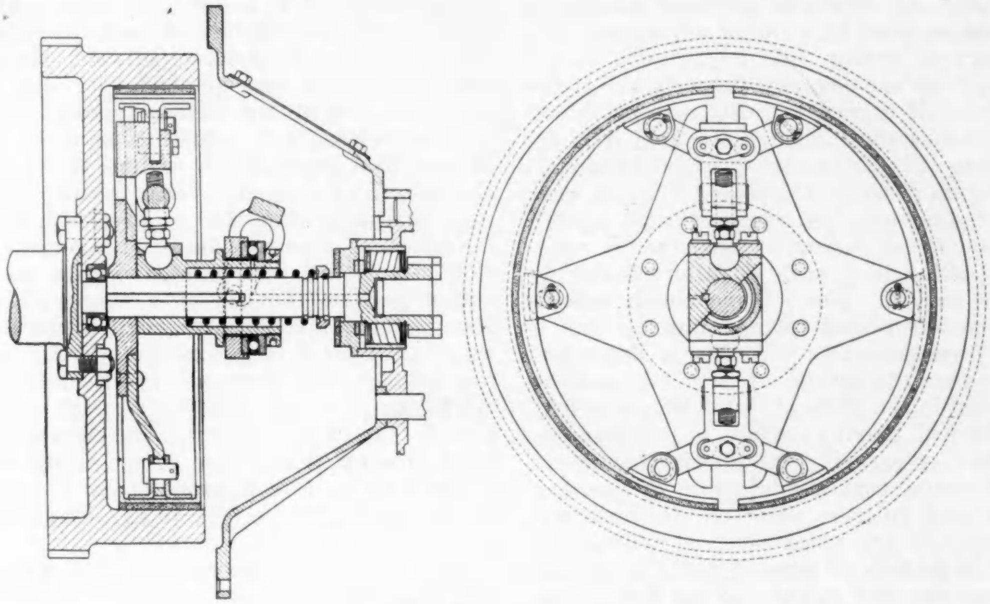


FIG. 28—THE PFEIFFER SHOE-TYPE CLUTCH

when the clutch is disengaged. When radial loads are imposed, either radial or angular-contact ball-bearings are usually employed. These types are somewhat more expensive than the straight-thrust type, but are much better suited to carry radial loads; in fact, the straight-thrust type is seldom recommended, although it is sometimes used when the radial load is light. It is then apt to become noisy and wear eccentric grooves.

When the thrust-bearing is enclosed by a casing which is held against rotation by trunnion-pins bearing upon the throw-out lever, which is a construction that is employed frequently, the bearing runs continually when the clutch is engaged and the engine is turning over. Although the load under these conditions is light, the continual running, frequently at high speed, inevitably causes wear, even though lubrication is facilitated with this construction. A design preferred by many engineers is that in which the bearing does not run continuously. The throw-out levers or ring are arranged to clear the thrust-bearing except during the periods of disengagement, which are relatively very short as compared to the periods of engagement. When so arranged the bearing wears longer if properly lubricated, and even then requires but little lubricant. In fact, it is generally enclosed and packed with light grease which lasts in numerous cases for many months or even years of operation. In unit powerplants whose service is heavy and continuous, provision is sometimes made for lubrication by oil from the transmission by using a hollow clutch-shaft drilled at a point near the bearing. If desired, the pilot bearing can be lubricated in similar fashion, but means to prevent excess oil from reaching the clutch-facings are necessary; otherwise, in the dry type of clutch, slipping will occur. The pilot-bearing, usually carried in a recess in the flywheel or crankshaft flange, is frequently of the radial ball type, but plain or oilless bushings are often used. The pilot-bearing requires little lubrication, but is sometimes lubricated by oil seepage from a porous wicking-plug, one end of which is supplied with oil under pressure in the hollow crankshaft of the engine. The driven members of some clutches, especially those of the cone type, are carried on spigots or extensions of the crankshaft. Usually, they have plain bushes, but ball bearings are used in some cases, as will be seen by reference to the illustrations.

CLUTCH BRAKES

It is customary to provide a clutch brake to reduce the speed of rotation of the driven clutch-members on disengagement and facilitate gear-changing. This is usually in the form of a disc of small diameter arranged to bear on a flange or collar attached to the throw-out sleeve, and in general is made of fiber or asbestos composition. The brake disc is fastened to the gearbox as a rule, in the case of unit powerplants, or to the bell-housing when a separate gearset is employed, but it is sometimes allowed to float on the clutch-shaft between the thrust-bearing and some flat-faced stationary member. The clutch brake is of course more important with clutches having heavy driven-members with considerable inertia than with lighter driven-members, and is not essential in some types, especially the multiple-disc type in which the floating driven-members are light and of small diameter and decelerate rapidly, due perhaps to a degree of dragging which is not, however, sufficient to cause the difficulties which a more positive dragging incurs. In plate and disc clutches it is necessary to provide means whereby the driven members shall be free to float in an axial direction on disengagement, even after some wear takes place. In some designs small relief-springs are arranged to give more positive disengagement, but these are not essential except perhaps in some designs in which the plate or disc runs in oil, which may cause adhesion under certain conditions.

Some details of spring construction have already been discussed, but a few items in this connection remain to be considered. The space allowed for springs and the desirability of compactness in design make it necessary in most cases to use rather short springs. These are often too stiff; that is, for a small change in length, the pressure-change is considerable. The result is that wear on the friction surface decreases the pressure enough to cause slippage. This condition can often be bettered by a change of spring-section and material without variation in overall dimensions or, in other words, by an intelligent design of the spring. In many cases the springs

are mounted so that the decreased pressure occasioned by wear can be compensated by a simple adjustment but, with the direct-acting spring, this adjustment is not necessary if the springs are designed properly and if the capacity of the clutch is correct in relation to maximum engine torque. When several springs are used it is desirable to have them of uniform strength and flexibility, adjusted to give equal pressure to prevent unequal wear and cocking or binding upon release, and with smooth action on engagement, but reasonably satisfactory operation is sometimes obtained in spite of failure to comply closely with these general rules. Some clutch makers prefer a single centrally placed spring, claiming that it gives a more uniform pressure; but this is dependent upon other factors than the spring. This is indicated by the construction used in the Hilliard clutch shown in Fig. 24, wherein a spherical bearing-surface is provided between the pressure-ring carrying the multiplying levers and the throw-out sleeve, with the intention of insuring equal pressure at each fulcrum point on the pressure-plate. Multiple springs are often recessed within the flywheel or placed in pockets of pressed metal in the annular space between the disc facing and the hub. This gives a neat external appearance, reduces windage, facilitates enclosure and minimizes danger of injury when the clutch is turning.

Notes on the Theory of Design

It is not my intention to discuss the theory of clutch design at length, but rather to comment briefly on certain simple basic factors which must be well understood before an intelligent design can be laid down. The maximum torque-capacity of any clutch is given by the equation

$$T = PNrf$$

where

T = the maximum torque capacity in pound-feet
P = the total pressure on the friction surface in pounds
N = the number of engaging surfaces
r = the mean radius of the friction surface in feet
f = the coefficient of friction

Concerning the pressure factor, it is possible theoretically to build a clutch of any desired capacity by simply increasing the pressure on the friction surface, but there are practical limits beyond which it is impracticable to go. A 60-lb. pedal or throw-out pressure is about the maximum permissible, except perhaps in certain truck applications. For comfort in driving, 30 lb. is considered the permissible maximum pressure by some designers, while lighter pressures are desirable in cars intended for ladies' use. It is general practice among American car and transmission manufacturers to use a pedal measuring about 11 in. on the long arm and from $1\frac{1}{2}$ to $2\frac{1}{2}$ in. on the short arm which actuates the clutch. This gives an average leverage of 5.5 to 1. With this ratio and a maximum pedal pressure of 60 lb., the pressure on the short end would be 330 lb., but this pressure should preferably not exceed 165 lb. A compound reduction by links and levers or cams external to the clutch can be employed, but this involves extra parts which are not required otherwise. If direct-acting clutch-springs are to be employed, a pressure of 165 lb. is not sufficient to prevent slippage with the average engine unless the number of friction surfaces or the mean radius of the surface is increased unduly. Two alternatives are employed, so far as the clutch proper is concerned, (*a*) the use of levers to increase the pressure on the friction surface and (*b*) the use of wedging action to increase the pressure on the friction surface. The advantage and disadvantage of each method has been pointed out in the discussion of the various types of construction.

By reference to the illustrations and to Table 1 it will be seen that many clutch designs in which direct-acting springs are employed involve the use of much greater total pressures than 165 or even 330 lb. To decrease the pressure required for disengagement, levers of various forms are incorporated within the clutch. This makes possible the use of a single plate or of fewer discs without involving the chief disadvantage of levers which multiply spring-pressure. From the foregoing it will be seen that there are several methods of varying the factor P in the capacity formula and thus varying directly the clutch capacity, but it should be borne in mind however that, other things being equal, the wear on the friction face is proportional to the unit pressure on the surface; hence it is an advantage to keep the unit pressure low.

An inspection of the capacity formula shows immediately that the capacity of a clutch is directly proportional to the number of friction surfaces engaging. When floating discs are employed between driving and driven members, slip is divided between the two faces; hence, but one face is considered in determining the factor N in the formula of capacity. It is possible to double the capacity of a single-plate clutch by simply adding a second plate and a friction ring. The advantage of the multiple-disc type in this respect is obvious. In this type it is possible to adapt the same clutch to engines of different capacity readily by merely changing the number of plates. The number of plates used will determine, at least to some extent, the ease of engagement or how gradually the full load is picked up upon engagement of the clutch. Increasing the number of plates and decreasing the spring pressure increases the "softness" of engagement and facilitates starting without sudden shock or undue strain on the mechanism.

For practical purposes the mean radius of the friction surface can be taken as equal to one-half the sum of the maximum and minimum radii. The torque transmitted obviously increases as the distance from the center to the point of application of pressure increases, and in direct proportion to this distance; hence, a large clutch diameter is desirable so long as the inertia of the driven member is not too great. The maximum diameter is controlled, however, largely by the size of the flywheel and by the cost of construction, which increases considerably with size. For a given outside diameter of friction surface, the mean radius can be increased by using a narrow facing, especially on plate and disc clutches. Although this decreases the area of surface, practical experience has shown that, within certain limits, a narrower facing wears longer than a wide facing. This apparently is because of the greater uniformity of wear over the entire surface, due no doubt to the fact that the rubbing speed at the outer diameter of a wide annular surface is much greater than at the inner diameter.

Since by definition the coefficient of friction between two surfaces is the ratio obtained by dividing the force required to slide one surface over the other by the pressure on the surface, it is evident that, as applied to clutches, the coefficient is the ratio of the maximum torque each surface can carry to the total pressure on that surface. The coefficient varies greatly with the

TABLE 1—SPECIFICATIONS OF VARIOUS CLUTCHES

Cone Type

Figure Number	Name, Model and Nationality of Car, Truck or Clutch[4]	Number of Cylinders, Bore and Stroke, in.	Maximum Engine Torque, lb. ft.	Dry or in Oil	Facing Material[7]	Maximum Diameter of Facing, in.	Minimum Diameter of Facing, in.	Width of Friction Face, in.	Angle of Cone, deg.	Area of each Friction Face, sq. in.[8]	Number of Wearing Faces of Friction Material	Total Area of Wearing Faces of Friction Material sq. in.[8]	Mean Radius of Friction Face, in.[9]	No. of Driving Members[10]	No. of Driven Members[11]	Number of Springs[12]	Total Spring Pressure, lb.[13]	Total Pressure on Friction Face, lb.[14]	Pressure per Sq. In. of Friction Surface, lb[14]	Torque Capacity of Clutch when New, lb.-ft.[15]	Ratio of Maximum Torque Capacity of Clutch to Maximum Torque of Engine	Pedal Pressure Required to Disengage, lb.[18]	Drive Taken by[19]	Are Multiplying Levers or Toggle Used?[20]	Means of Adjustment[21]
4	Clement Talbot, 15 Hp. (E)	4-3 1/8x5 1/8	.....	Dry	Fabric	12 7/8	11.82	2.28	11	88.00	1	88.00	6.17	1	1	1	400	2,100	23.90	......	......	....	Univ.	No	None
..	Clement Talbot, 25 Hp. (E)	4-4 x5 1/2	.....	Dry	Fabric	14 9/16	13 3/8	2.28	11	101.00	1	101.00	6.98	1	1	1	500	2,630	26.00	......	......	....	Univ.	No	None
..	Commer, 2 Ton (E)	4-4 x4 3/4	.....	Dry	Leather	15 1/2	14 1/2	2.00	14	117.70	1	117.70	7 1/2	1	1	1	130	537	4.60	126	......	24	....	No	Spring Bolts
..	Commer, 3 1/2-5 Ton (E)	4-4 1/2x5 1/2	.....	Dry	Leather	17 1/8	16 1/8	2 1/2	11	130.60	1	130.60	8 5/16	1	1	1	220	1,160	8.90	220	......	40	Univ.	No	Spring Bolts
..	Dennis, 3-4 Ton (E)	4-4 1/2x5 15/16	.....	Dry	Fabric	17 3/8	15 7/8	3 1/2	12	183.00	1	183.00	8 5/16	1	1	(3)	285	1,380	7.50	208	......	58	Univ.	No	Spring Bolts
..	Guy, (P) 20 Hp. (E)	8-2 13/16x4 15/16	.....	Dry	Fabric	15 3/8	14 3/8	2 3/8	12	111.00	1	111.00	7 7/16	1	1	(4)	160	770	6.90	......	......	35	Univ.	No	Spring Bolts
..	Guy, 2 1/2 Ton (E)	4-4 x5 1/2	.....	Dry	Fabric	17 5/16	15 7/8	2 1/2	16	130.60	1	130.60	8.30	1	1	(4)	160	580	4.50	......	......	60	....	No	Spring Bolts
..	Hartford, (C) (A)	....	.....	Dry	Leather	15 3/4	14 5/8	2 5/8	12 1/2	125.00	1	125.00	7 19/32	1	1	1	225 to 350	1,040 to 1,620	8.30 to 12.90	......	......	....	Univ.	No	Adjustable Spring
3	Humber, 10 Hp. (E)	4-2 9/16x4 3/4	.....	Oil	Leather	9 7/8	9 7/16	1.77	14	55.50	1	55.50	4.83	1	1	1	280	1,155	21.00	50	......	30	Keys	No	Spring Nut
..	Humber, 16 Hp. (E)	4-3 1/8x5 1/2	.....	Oil	Leather	11.82	11 1/4	1.58	14	57.20	1	57.20	5 21/32	1	1	1	245	1,015	17.70	92	......	30	Univ.	No	....
..	Maudesley, 3 Ton (E)	4-4 1/2x5	.....	Dry	Leather	19 7/8	18 3/4	2 9/16	12 1/2	155.00	1	155.00	9 21/32	1	1	1	600	2,775	17.90	......	......	25	Univ.	No	None
..	Oakland, (P) (A)	6-2 13/16x4 3/4	.....	Dry	Leather	12 9/32	.....	1 1/2	11	57.00	1	57.00	6.00	1	1	(4)	260	1,370	24.00	137	1.3	15	....	No	None
..	Sunbeam, 16 Hp. (E)	4-3 1/8x5 15/16	.....	Dry	Leather	14 1/8	13 1/16	2 3/8	12	101.00	1	101.00	6.80	1	1	(4)	300	1,450	14.40	100	......	50	Univ.	No	Spring Bolts
5	Thornycroft, 3 Ton (E)	4-4 1/2x6	.....	Dry	Fabric	18 7/8	17 5/8	3.00	10	172.20	1	172.20	8 9/16	1	1	2	300	1,730	10.00	375	......	18	Univ.	No	None
..	Vulcan, 16 Hp. (E)	4-3 1/8x5 1/8	.....	Dry	Fabric	14 3/4	13 3/8	2 3/4	15	121.50	1	121.50	7.03	1	1	(3)	290	1,120	9.30	186	......	40	Univ.	No	Spring Bolts

Single-Plate Type

Figure Number	Name, Model and Nationality of Car, Truck or Clutch[4]	Number of Cylinders, Bore and Stroke, in.	Maximum Engine Torque, lb. ft.	Dry or in Oil	Facing Material[7]	Maximum Diameter of Facing, in.	Minimum Diameter of Facing, in.	Width of Friction Face, in.	Angle of Cone, deg.	Area of each Friction Face, sq. in.[8]	Number of Wearing Faces of Friction Material	Total Area of Wearing Faces of Friction Material sq. in.[8]	Mean Radius of Friction Face, in.[9]	No. of Driving Members[10]	No. of Driven Members[11]	Number of Springs[12]	Total Spring Pressure, lb.[13]	Total Pressure on Friction Face, lb.[14]	Pressure per Sq. In. of Friction Surface, lb[14]	Torque Capacity of Clutch when New, lb.-ft.[15]	Ratio of Maximum Torque Capacity of Clutch to Maximum Torque of Engine	Pedal Pressure Required to Disengage, lb.[18]	Drive Taken by[19]	Are Multiplying Levers or Toggle Used?[20]	Means of Adjustment[21]
13	Arrol Johnston (E)	4-3 1/8x4 3/4	.....	Dry	Fabric	8 3/8	4 11/16	1 27/32	....	37.80	4	151.20	3.26	2	1	3	900	900	23.80	100	......	60	Splines	Yes	Spring Caps
12	Austin, 20 Hp. (E)	4-3 3/4x5	.....	Dry	Fabric	11 7/8	8 1/2	1 11/16	....	54.00	2	108.00	5 3/16	2	1	3	900	900	16.70	......	......	28	Splines	Yes	None
6	Autocar, XXI-F (T) (A)	2-4 3/4x4 1/2	90	Dry	Molded	12 3/4	9 1/2	1 5/8	....	56.80	2	113.60	5 9/16	2	1	8	800	800	14.10	187.5	2.08	50	Keys and Gear Teeth	Yes	Screws on Levers
..	Autocar, XXVI-Y (T) (A)	4-4 1/4x5 1/2	185	Dry	Molded	12 3/4	9 1/2	1 5/8	....	56.80	2	113.60	5 9/16	2	1	8	1,440	1,440	25.30	338	1.83	50	Keys and Gear Teeth	Yes	Screws on Levers
..	Borg & Beck, M (C) (A)	....	.....	Dry	Fabric	7 3/4	5 1/4	1 1/4	....	25.00	4	100.00	3 1/4	2	1	1	225 to 325	.....	80 to 110	80 to 100	......	....	Keys and Splines	Yes	
..	Borg & Beck, DX (C) (A)	....	.....	Dry	Fabric	9 5/8	6 3/4	1 1/2	....	38.00	4	152.00	4 1/16	2	1	1	225 to 325	.....	50 to 70	120 to 170	......	....	Keys and Splines	Yes	
..	Borg & Beck, GX (C) (A)	....	.....	Dry	Fabric	11 3/4	8 1/4	1 11/16	....	52.00	4	208.00	4 29/32	2	1	1	225 to 375	.....	35 to 55	150 to 240	......	....	Keys and Splines	Yes	Rotation of Toggle Mounting Ring
..	Borg & Beck, RGX (C) (A)	....	.....	Dry	Fabric	11 3/4	7 7/8	2 1/4	....	67.00	4	268.00	4 3/4	2	1	1	225 to 375	.....	25 to 40	150 to 240	......	....	Keys and Splines	Yes	
14	Borg & Beck, JX (C) (A)	....	.....	Dry	Fabric	13 3/4	7 7/8	3	....	101.00	4	404.00	5 3/8	2	1	1	225 to 375	.....	20 to 30	210 to 310	......	....	Keys and Splines	Yes	

Single-Plate Type (Concluded)

Figure Number	Name, Model and Nationality of Car, Truck or Clutch[4]	Number of Cylinders Bore and Stroke, in.	Maximum Engine Torque, lb.-ft.	Dry or in Oil	Facing Material[7]	Maximum Diameter of Facing, in.	Minimum Diameter of Facing, in.	Width of Friction Face, in.	Angle of Cone, deg.	Area of each Friction Face, sq. in.[8]	Number of Wearing Faces of Friction Material	Total Area of Wearing Faces of Friction Material sq. in.[8]	Mean Radius of Friction Face, in.[9]	No. of Driving Members[10]	No. of Driven Members[11]	Number of Springs[12]	Total Spring Pressure, lb.[13]	Total Pressure on Friction Face, lb.[14]	Pressure per Sq. In. of Friction Surface, lb.[14]	Torque Capacity of Clutch when New, lb.-ft.[15]	Ratio of Maximum Torque Capacity of Clutch to Maximum Torque of Engine	Pedal Pressure Required to Disengage, lb.[18]	Drive Taken by[19]	Are Multiplying Levers or Toggles Used?[20]	Means of Adjustment[21]
9	Bristol, 4 Ton (E)	4-4 1/2x3 3/4	...	Dry	...	11 3/8	8 1/4	1 9/16	...	48.16	2	96.32	4 29/32	2	1	(3)	1,000	1,000	20.70	...	...	35	Spline	Yes	Screw on Toggle Fulcrum
7	Dennis, 2 Ton (E)	4-4 1/8x5 15/16	...	Dry	Fabric	13	8 7/16	2 9/32	...	76.83	2	153.66	5.36	2	1	(3)	1,236	1,236	8.90	16	...	64	Univ.	Yes	Screws on Toggle Levers
10	Halley, 3 1/2 Ton (E)	6-3 1/2x6	...	Dry	Fabric	15 1/8	11 3/16	1 31/32	...	81.50	2	163.00	6.57	2	1	(3)	1,230	1,230	15.10	238	...	53	Spline	Yes	None
..	Hoosier, K1 (C) (A)	...	...	Dry	M or F	7 13/16	5 7/16	1 3/4	...	25.00	4	100.00	3 5/16	2	1	1	400	...	252.00 Max	120	...	...	Keys and Splines	Yes	Threaded Ring
15	Hoosier, K5 (C) (A)	...	...	Dry	M or F	9 7/8	6 13/16	1 1/2	...	40.20	4	161.00	4.17	2	1	1	400	...	60.00 Max	200	...	...	Keys and Splines	Yes	Threaded Ring
15	Hoosier, K6 (C) (A)	...	...	Dry	M or F	11 7/8	8 1/8	1 3/4	...	58.90	4	236.00	5.00	2	1	1	400	...	70.00 Max	300	...	...	Keys and Splines	Yes	Threaded Ring
..	Lauth, (C) (A)	...	...	Dry	...	13 1/4	9 1/4	2	...	70.00	4	280.00	5 5/8	2	1	1	...	...	...	...	...	...	Keys and Splines	Yes	...
11	Mack, AC (T) (A)	4-5 x6	250	Dry	Fabric	19	13	3	...	151.00	2	302.00	8.00	2	1	9	1,035	1,035	6.90	...	...	...	Splines	Yes	None
8	Napier, 40-50 Hp. (E)	...	...	Dry	...	10 7/16	6 15/16	1 3/4	...	47.80	4	191.20	4 11/32	2	1	(3)	...	...	...	...	...	...	Splines	Yes	Screws on Levers
16	Rover, 12 Hp. (E)	4-2 15/16x5 1/8	...	Oil	Bronze	10 5/8	6 3/8	2 1/8	...	57.60	2	115.20	4 9/32	2	1	1	...	1,740	30.20	90	...	30	Univ.	Yes	Spring Caps
17	Twin Disc, A (C) (A)	...	...	Dry	Either	11 1/2	5 1/2	3	...	80.10	2	160.20	4 1/4	1	2	None	...	...	25 to35	456	1.83[17]	...	Pins	Yes	Threaded Lever Yoke

Multiple-Disc Type

Figure Number	Name, Model and Nationality of Car, Truck or Clutch[4]	Number of Cylinders Bore and Stroke, in.	Maximum Engine Torque, lb.-ft.	Dry or in Oil	Facing Material[7]	Maximum Diameter of Facing, in.	Minimum Diameter of Facing, in.	Width of Friction Face, in.	Angle of Cone, deg.	Area of each Friction Face, sq. in.[8]	Number of Wearing Faces of Friction Material	Total Area of Wearing Faces of Friction Material sq. in.[8]	Mean Radius of Friction Face, in.[9]	No. of Driving Members[10]	No. of Driven Members[11]	Number of Springs[12]	Total Spring Pressure, lb.[13]	Total Pressure on Friction Face, lb.[14]	Pressure per Sq. In. of Friction Surface, lb.[14]	Torque Capacity of Clutch when New, lb.-ft.[15]	Ratio of Maximum Torque Capacity of Clutch to Maximum Torque of Engine	Pedal Pressure Required to Disengage, lb.[18]	Drive Taken by[19]	Are Multiplying Levers or Toggles Used?[20]	Means of Adjustment[21]
..	Browne, (C) (A)	...	...	Dry	Molded	7 7/8	6 1/8	7/8	...	19.20	6 to 14	115 to 268	3 1/2	3 to 7	4 to 8	3	450	450	23.40	275 to 640	1.67[17]	...	Teeth and Lugs	Yes	Screws in Levers
	Detlaff, H (C) (A)	...	...	Dry	M or F	8 3/8	6 1/2	15/16	...	22.00	6 to 18	132 to 396	3 23/32	3 to 9	4 to 10	3	300 to 375	300 to 375	14 to 17	249 to 747	2.50[17]	...	Gear Teeth	No	Spring Bolt
20	Detlaff, L & J (C) (A)	...	...	Dry	M or F	7 7/8	5 3/8	1 1/4	...	26.00	6 or 8	156 to 208	3 5/16	3 or 4	4 or 5	3	300 to 375	300 to 375	11 1/2 to 14 1/2	225 to 300	2.50[17]	...	Pins	No	None
..	Essex, (P) (A)	4-3 3/8x5	...	Oil	Cork	...	...	...	...	...	8	...	...	4	5	3	...	...	...	...	...	...	Pins and Gear Teeth	No	...
..	Fuller, GCL (C) (A)	...	...	Dry	M or F	8	6	1	...	22.00	14	308.00	3.50	7	8	1	350	350	15.90	196	...	...	Pins	No	None
23	G. M. C., (T) (A)	4-4 1/2x6	203.0	Dry	Molded	8 3/8	6 1/4	1 1/16	...	14.50	16	232.00	3.66	8	9	2	320	320	22.00	...	...	33	Gear Teeth and Keys	No	None
24	Hilliard, XDA (C) (A)	...	...	Either	Molded	10 3/4	6 3/4	2	...	55.00	4	220.00	4 3/8	3	2	1	380	1,900	34.50	630 or 787[16]	...	...	Gear Teeth	Yes	Screws on Cover-Plate
..	Hilliard, S-6 (C) (A)	...	...	Either	Molded	12	8	2	...	63.00	6	378.00	5.00	4	3	1	375	1,875	30.00	709 or 997[16]	...	...	Gear Teeth	Yes	Screws on Cover-Plate
..	Hillard, S-8 (C) (A)	...	...	Either	Molded	12	8	2	...	63.00	8	504.00	5.00	5	4	1	375	1,875	30.00	1,155 or 1,418[16]	...	...	Gear Teeth	Yes	Screws on Cover-Plate
..	Hoosier, K19 (C) (A)	...	...	Either	M or F	7 1/8	5 3/8	7/8	...	17.00	4	68.00	3 1/8	3	2	1	400	...	...	205	...	...	Keys and Splines	Yes	Threaded Ring
..	Hoosier, K20 (C) (A)	...	...	Either	M or F	9	6 1/2	1 1/4	...	30.00	4	120.00	3 7/8	3	2	1	400	...	...	300	...	...	Keys and Splines	Yes	Threaded Ring
..	Hudson, (P) (A)	6-3 1/2x5	...	Oil	Cork	...	...	...	...	...	16	...	...	8	8	1	...	...	...	...	...	...	Pins and Gear Teeth	No	...
19	Hupmobile, (P) (A)	4-3 1/4x5 1/2	111.5	Dry	Fabric	9 5/8	7 3/8	1 1/8	...	30.80	8	246.40	4 1/4	4	5	6	270	270	8.77	...	...	...	Pins	No	Notched Spring Bolt

Multiple-Disc Type (Concluded)

Figure Number	Name, Model and Nationality of Car, Truck or Clutch[4]	Number of Cylinders Bore and Stroke, in.	Maximum Engine Torque, lb.-ft.	Dry or in Oil	Facing Material[7]	Maximum Diameter of Facing, in.	Minimum Diameter of Facing, in.	Width of Friction Face, in.	Angle of Cone, deg.	Area of each Friction Face, sq. in.[8]	Number of Wearing Faces of Friction Material	Total Area of Wearing Faces of Friction Material sq. in.[8]	Mean Radius of Friction Face, in.[9]	No. of Driving Members[10]	No. of Driven Members[11]	Number of Springs[12]	Total Spring Pressure, lb.[13]	Total Pressure on Friction Face, lb.[14]	Pressure per Sq. In. of Friction Surface, lb.[14]	Torque Capacity of Clutch when New, lb.-ft.[15]	Ratio of Maximum Torque Capacity of Clutch to Maximum Torque of Engine	Pedal Pressure Required to Disengage, lb.[18]	Drive Taken by[19]	Are Multiplying Levers or Toggles Used?[20]	Means of Adjustment[21]
..	Lexington, (P) (A)	6-3¼x4½	.....	Dry	....	.....	.....	.....	....	......	.....	....	....	...	...	.....	270	270	....	....	....	....	Gear Teeth	....	....
25	Locomobile, (P) (A)	6-4½x5½	283.0	Dry	Molded	10¾	8¼	1¼	....	37.30	10	373.00	4¾	5	5	1	285	285	7.64	356	1.28	....	Keys	No	None
..	Mack, AC (T) (A)	4-4 x5	133.0	Dry	Fabric	$8\frac{7}{16}$	6¼	$1\frac{3}{32}$	....	25.10	12	301.20	3.68	6	7	2	320 to 340	320 to 340	12.80 to 13.60	....	....	....	Keys and Gear Teeth	No	Spring Bolts
..	M. & E., 8U (C) (A)	....	.....	Dry	Fabric	$7\frac{3}{16}$	$5\frac{3}{16}$	1	....	19.50	4	78.00	$3\frac{3}{32}$	2	2	1	100	463	24.00	... 230	....	....	Keys and Gear Teeth	Yes	Screws on Cover-Plate
26	M. & E., 10U (C) (A)	....	.....	Dry	Fabric	$9\frac{3}{16}$	$6\frac{5}{16}$	$1\frac{7}{16}$	....	35.00	4	140.00	3⅞	2	2	1	150	665	19.00	... 500	....	....	Keys and Gear Teeth	Yes	Screws on Cover-Plate
..	M. & E., 12U (C) (A)	....	.....	Dry	Fabric	$11\frac{3}{16}$	$7\frac{3}{16}$	2	....	57.70	4	231.00	$4\frac{19}{32}$	2	2	1	200	925	16.00	... 700	....	....	Keys and Gear Teeth	Yes	Screws on Cover-Plate
..	M. & E., 12HD (C) (A)	....	.....	Dry	Fabric	$11\frac{3}{16}$	$7\frac{3}{16}$	2	....	57.70	6	346.00	$4\frac{19}{32}$	3	3	1	250	1,000	19.00	.. 1,000	....	....	Keys and Gear Teeth	Yes	Screws on Cover-Plate
21	Packard, (P) Single Six (A)	6-3⅜x4½	140.0	Dry	Molded	8	6	1	....	22.00	6	132.00	3½	4	3	8	400	400	18.20	....	....	20	Gear Teeth	No	Spring Bolts
..	Packard, (P) Twin Six (A)	12-3 x5	238.0	Dry	Molded	8	6	1	....	22.00	12	264.00	3½	6	7	1	400	400	18.20	....	....	22	Keys	No	Spring Bolts
..	Packard, 2 Ton (T) (A)	$4\text{-}4\frac{3}{16}$x5½	168.0	Dry	Molded	8	6	1	....	22.00	10	220.00	3½	5	6	1	490	490	22.30	....	....	25	Keys	No	None
..	Packard, 3-5 Ton (T) (A)	4-5 x5½	210.0[5]	Dry	Molded	8	6	1	....	22.00	14	308.00	3½	7	8	1	490	490	22.30	....	....	25	Keys	No	None
22	Reo, (P) (A)	$6\text{-}3\frac{3}{16}$x5	238.0[6]	Dry	Fabric	6⅝	5⅛	....	....	13.84	13	180.00	2.94	7	6	3	240	240	17.30	....	....	....	Gear Teeth	No	None
18	Vauxhall, (P) (E)	4-3¾x5½	.....	Oil	Bronze	8½	$7\frac{7}{16}$	$\frac{17}{32}$	....	13.34	22	294.00	3.98	11	12	1	90	90	6.70	150	....	17	Gear Teeth	No	None
..	Warner, K19 (C) (A)	....	.....	Dry	....	.....	.....	....	....	20.05	8 to 18	164 to 369	....	4 to 9	5 to 10	3	270 to 405	270 to 405	13.20 to 19.70	140 to 315	....	....	Gear Teeth	No	Spring Bolts

Shoe Type

Figure Number	Name, Model and Nationality of Car, Truck or Clutch	Number of Cylinders Bore and Stroke, in.	Maximum Engine Torque, lb.-ft.	Dry or in Oil	Facing Material	Maximum Diameter of Facing, in.	Minimum Diameter of Facing, in.	Width of Friction Face, in.	Angle of Cone, deg.	Area of each Friction Face, sq. in.	Number of Wearing Faces of Friction Material	Total Area of Wearing Faces of Friction Material sq. in.	Mean Radius of Friction Face, in.	No. of Driving Members	No. of Driven Members	Number of Springs	Total Spring Pressure, lb.	Total Pressure on Friction Face, lb.	Pressure per Sq. In. of Friction Surface, lb.	Torque Capacity of Clutch when New, lb.-ft.	Ratio of Maximum Torque Capacity of Clutch to Maximum Torque of Engine	Pedal Pressure Required to Disengage, lb.	Drive Taken by	Are Multiplying Levers or Toggles Used?	Means of Adjustment
..	Pfeiffer, (C) (A)	....	.....	Either	Fabric	13	13	2½	....	49.50	2	99.00	6.50	1	2	1	300	......	....	....	......	....	Pins and Rivets	Yes	Lengthen Toggle

[4] (*A*)—American, (*C*)—Clutch, (*E*)—English, (*P*)—Passenger Car and (*T*)—Truck.
[5] Figures are for 3-ton truck.
[6] Figures are for 5-ton truck.
[7] Bronze indicates bronze plate without facing material, *M*—Molded asbestos composition and *F*—Woven asbestos fabric.
[8] Areas given are computed from the diameter and width without any allowance for rivet holes or depressions for other holding devices.
[9] Figures given are the arithmatical mean radii.
[10] Figures given include the flywheel or members attached rigidly thereto, taken as one member when the friction face bears upon the flywheel or the rigidly connected member.
[11] Figures include only members which transmit torque directly from driving members; a driven member made up of two or more pieces is considered as one member, and driven members such as levers and throw-out sleeves which do not transmit torque *directly* from driving members are not included.
[12] Figures in parentheses are uncertain.
[13] When two figures are given, the clutch maker furnishes springs giving either pressure, or some intermediate pressure.
[14] Figures are presumed to be the maximum pressures when the clutch is new and properly adjusted. When toggles are used to multiply the spring pressure, any slight alteration in the position of the toggles, due to wear or lack of adjustment can make marked variation in total or unit pressure.
[15] In some cases these figures indicate the actual or computed maximum torque capacity, and in others "safe" or "recommended" capacity. In some cases the manufacturer makes clutches of different capacity than the model listed, or will vary the capacity to suit conditions by a variation in the number of discs, friction material or spring pressure.
[16] The first figures are the capacity when arranged to run in oil and the others are the capacity when used dry.
[17] This figure corresponds to the manufacturer's recommendation. Other figures in this column are actual ratios employed presumably with the clutch new or in perfect adjustment.
[18] Figures in this column are the actual pressures used or reported used by the respective manufacturers.
[19] Univ.—Universal-joint. In nearly all clutches the driven members, and in some types the driving members also, must be allowed a certain free axial motion upon release, and some member such as keys, splines, gear teeth or pins which transmit torque and at the same time allow axial motion are therefore provided and these members are listed in this column. The pin or block of the universal-joint frequently performs this function, especially in the case of cone clutches.
[20] Data in this column refer to levers which rotate with and form a part of the clutch, and not to non-rotating throw-out levers external to the clutch.
[21] Data in this column refer to adjusting means within the clutch. External adjusting means are often provided but not referred to here.

nature of the surface and its condition, especially with respect to lubrication. Between smooth metal surfaces wetted with oil, it is about 0.07. Woven asbestos fabric on smooth steel gives an average coefficient of about 0.3, but this varies considerably with wear, the temperature and the nature of the impregnating material used in the manufacture of the fabric. If the impregnating material contains parafin or other substances which, under the influence of heat, act to some extent as lubricants, the coefficient will decrease as the temperature rises, so that a clutch which performs well when cool sometimes starts suddenly to slip when it has become heated by the slipping that always occurs on engagement. Frequently, repeated engagements such as occur when driving in traffic or careless handling during periods when the clutch is intentionally allowed to slip, sometimes raise the temperature to 500 deg. fahr. or higher; hence, the facing material must be capable of withstanding this temperature and the factor of safety as regards capacity should be sufficient to permit operation even with the reduced coefficient at high temperature. The friction coefficient of woven linings is known to vary through a wide range. One engineer who has made tests with woven facings reports coefficients varying from 0.27 to 0.38. It is considered good practice to use a clutch that has a normal torque capacity when new of 1.6 to 2 times the maximum torque of the engine, although some engineers state that smoother action results when the torque capacity of the clutch is only slightly in excess of that of the engine.

A clutch that will carry a given maximum torque indefinitely when once fully engaged will not always pick up this load without slipping for a relatively long period during which excessive heat is generated. One investigator states that experiment has shown, in the case of multiple-disc clutches at least, that a desirable softness of engagement is secured without undue heating by a clutch that will slip from 50 to 60 revolutions when suddenly assuming full load, and that this slip is obtained when the maximum load carried averages 70 per cent of the total load that the clutch will carry without slip when fully engaged. Within the last two or three years, facings of molded asbestos composition have come to be used widely. They are said to have a friction coefficient of 0.5 and to be more durable than other facing materials. They are of uniform texture, can be made in quantity to close tolerances and do not require a joint secured with wire staples such as is generally employed with the woven fabric. It should be noted that the coefficient of friction is independent of the area of contact surface. For this reason a narrow facing under a given pressure will carry the same torque as a wide facing having the same mean radius, but the unit pressure will of course be greater on the narrower facing. This will tend to cause greater wear, but the wider surface, as pointed out, does not always wear longer because of the greater difference in speed at inner and outer diameters. The ideal condition in this respect is approached most closely by the use of a number of relatively narrow surfaces, thus giving a large total area, a consequent long life and a more gradual picking up of the load, without causing wide differences in speed between the various parts of the contact surface.

In Conclusion

From the foregoing analysis of factors entering into the capacity formula, it will be seen that torque capacity can be made to vary by changes in each of the four factors involved. In other words, the same torque can be obtained by a great number of combinations. The best combination is that which gives the longest life, requires the least attention, is the smoothest in engagement, has the lowest inertia and is the least expensive to manufacture. For passenger-car use at least, the multiple-disc type seems to fill best the greatest number of these conditions, but it is evident from the number of other types used that engineers are far from being agreed on this point. In this connection it is worthy of note that there is wide variation in multiple-disc clutch design. When the price must be kept as low as possible, the number of plates is usually reduced to the minimum and the pressure on the discs is increased; but when cost is secondary to the smoothest action on engagement, the number of plates is increased and the pressure reduced, at the same time prolonging the life of the facings by adding to the total wearing surface.

The data given in Table 1 have been gathered from a variety of sources and no claim is made for their absolute accuracy. Many of the dimension figures were obtained by scaling blueprints and, consequently, they may vary slightly from the actual dimensions used. It will be seen, however, that practice varies so widely in many particulars that the comparisons afforded by the figures given are close enough for practical purposes. It is my hope and belief that the data and drawings collected here will prove useful to engineers who are called upon to design clutches or to make a selection for a given purpose from among the various types of clutch available.

FLIGHTS OF TWO NEW AIRPLANES AT McCOOK FIELD

Two new airplanes built for the Air Service were given initial flights at McCook Field, Dayton, Ohio, recently. The Loening PW-2, single-seater pursuit airplane was flown by Lieut. J. A. Macready, who reported that it handled very well, was pleasant to fly and possessed excellent visibility. He experienced no difficulties on the flight, which lasted about ½ hr. The airplane, which is equipped with a 300-hp. Wright engine with a four-bladed propeller and carried the full military equipment for the single-seater pursuit airplane developed during the war, is a monoplane with the wings attached to the upper longerons and braced by diagonal struts to the lower longerons of the fuselage.

The G. Elias TA-1 two-seater training airplane was flown by Lieut. George B. Patterson who reported that the airplane balanced perfectly and seemed very light and responsive to the controls. It appears to land very slowly and stops after an unusually short run. The machine is equipped with the 170-hp. Wasp ABC air-cooled radial engine and is the first military airplane that has been flown with the U. S. A. 27 wing-curve, which has given remarkable results in the wind-tunnel tests. With this wing-section only one pair of struts on each side is required for bracing, which greatly simplifies rigging and maintenance in the field. With the exception of the strut arrangement the general design of the airplane follows what is now considered orthodox practice.—Air Service News Letter.

Tractor Pulley Widths and Speeds

IN connection with the Farm Power Meeting of the Society held at Columbus, Ohio, Feb. 10, 1921, a report of the committee that has been invstigating tractor pulley widths and speeds was made by John Mainland, chief engineer of the thresher works of the Advance-Rumely Co., La Porte, Ind.

Committee Report

Our committee held a meeting on Oct. 4, 1920, at Racine, Wis., at which the entire personnel was present. There was a general discussion of the problems under consideration and the following decisions were made.

> The speeds already arrived at by a former committee, 1500, 2600, 3000 and 3500 ft. per min., were approved by the members of this committee.
>
> This committee recommends to designers of pulleys and clutches for new equipment that the minimum diameter be 12 in. and that the pulley width be not less than ½ in. wider than the belt required.
>
> It is the recommendation of the committee that a governor be considered a necessary part of a farm tractor that is used for belt operations and suggests that some means should be provided on tractors for the attachment of a suitable speed-indicating device.

The pulley speeds mentioned above are now published in the S. A. E. Handbook.

A motion was passed that 5, 6, 7, 8 and 9 in. should be a sufficient number of drive-belt widths for all tractor purposes. Chairman Mainland doubted whether a 5-in. width is necessary, but the committee included it, thinking that some of the small tractors might need a belt of this width.

The matter of a standard length for belts was discussed. No recommendation was made but some data were submitted by companies that furnish belts.

On page K40, Vol. 1, S. A. E. Handbook, issued in October, 1920, it is specified that tractors of from 10 to 20 hp. should have a pulley width of 4½ to 6½ in. In Mr. Mainland's opinion, from the standpoint of the tractor a 4 or 6-in. belt-width is perhaps sufficient to transmit the power at the correct belt speed of 3000 ft. per min., but in the field, from a practical standpoint, a tractor with a 4-in. belt is the height of folly so far as threshing is concerned. To put out a tractor with a 4-in. belt is not practical. Mr. Mainland said that he had never seen a separator that could be operated successfully with a 4-in. belt. There are many reasons why it is not practical. If one could set a tractor absolutely in line with a 4-in. pulley, perhaps it would work, but a tractor is operated in the field where the ground is not level and the tractor cannot be made level. For this reason, to operate successfully, it is necessary to have the pulleys more than ½ in. wider than the belt that is to be used.

Mr. Mainland stated at the committee meeting that the pulley on a tractor should be at least 2 in. wider than the belt necessary to transmit the power. Other members of the committee objected to that for various reasons, one of which was that while it is preferable to have the wider pulley there is not room for it in some cases. Mr. Mainland said that tractors run threshing machines more than any other type of machine. He continued: On a threshing machine that we build we make no pulley less than 9 in. wide and we used a 9-in. pulley with a 6-in. belt.

Governors

Several tractors have been put out that have no governors on them. In operating implements perhaps the governor is not necessary, but the tractor, to give its full value, should be able to operate all farm belt machinery. The cylinders of a threshing machine travel at a speed of 6250 ft. per min. on an average. This speed is greater than that recommended for flywheels on steam engines. We all know the old rule that a flywheel should not run at a surface speed of more than 1 mile per min. In operating a separator with a tractor that has no governor, if anything goes wrong and feeding is stopped, the tractor immediately runs away with the separator. It cannot do otherwise. With such a high normal speed, a speed is soon reached that will burst the cylinder of the threshing machine. It is a serious matter when a cylinder of a separator bursts during operation. Almost every State has passed laws in regard to flywheels and precautions have been taken to prevent flywheels from bursting. It is just as necessary to take similar precautions in the case of separator cylinders. If many tractors are marketed without governors, laws will be passed shortly to stop the practice. The Society should take steps to stop the practice before the States pass such laws.

Perhaps there is no subject of which less is known than that of belt speed and pulley widths under the varying conditions of farm work. A machine designed by the Sprague Electric Co. gives very accurate results on belt-driven machinery with regard to the pulleys. So far it has built three of these machines; the earliest one is in the possession of the Goodrich Co.; the second one is at Cornell University; the third and most advanced machine has just been installed at the plant of the Manhattan Rubber Co. I am in close touch with the chief engineer of the Manhattan Rubber Co. and he has told me that all information secured will be available for our committee.

The Discussion

Chairman E. A. Johnston:—If the tractor industry is to have a full measure of success, the tractor must be designed to operate profitably in the hands of the user. Belt slippage or any delay of the tractor outfit on account of trouble with belts is a serious loss.

With reference to the 4-in. pulley mentioned by Mr. Mainland, I believe that that was included because it was felt that for some of the lighter work, such as operating feed-grinders and some of the smaller machines, it would be sufficient. I agree fully with Mr. Mainland that a 4-in. belt is altogether too small to operate a separator economically. With reference to governors, I believe that most tractor designers appreciate fully the need for governors for controlling belt-driven machinery, such as ensilage cutters. There are one or two makes of tractor on which governors are omitted, which is unfortunate for the tractor industry as a whole.

INTERNAL-COMBUSTION ENGINE LUBRICANTS

IN the choice of lubricating oils for the internal-combustion engine the character of the oil used is mainly determined by the high temperatures produced in the cylinders by the combustion of the fuel; it is necessary that no marked decomposition of the oil shall occur to form carbon deposits which will interfere with the smooth and efficient operation of the engine. Minimum carbonization of the oil must therefore take place in the cylinders, and on the other hand the oil must reduce friction as far as possible and remain in effective service for a long time.

Lubricating Oil Tests

The nature of the oil used for lubrication depends on certain features of the engine, such as the method of cooling, engine speed and the clearances of the engine, and accordingly oils of different physical properties varying in viscosity and flash-point are required for the many types of engine. Certain qualities, however, should be possessed by such oils to insure that the engine will continue in serviceable operation without the necessity of frequent overhauling on account of carbon deposits and excessive wear. It is for this reason that tests other than those usually made on oils for the lubrication of machinery operating at ordinary temperatures are necessary to discriminate between various oils intended for the lubrication of internal-combustion engines.

These special laboratory tests have been devised to avoid the necessity of carrying out prolonged and expensive engine tests. Naturally it is of the greatest importance to insure that the results of these laboratory tests do actually correspond with those obtained when the same lubricating oils are tried out in engines; unfortunately, many difficulties arise in carrying out such tests on a sufficiently large scale to determine definitely the relative merits of various oils, since apparently minor factors in the operation of the tests may exert a far greater influence on the results than is produced by changing the lubricating oil. The special laboratory tests used for this purpose are (*a*) Evaporation loss; (*b*) carbonization test; (*c*) carbon content.

The evaporation loss and carbonization test are carried out by heating a weighed quantity of the oil to a high temperature, 300 to 550 deg. fahr., for a definite length of time; from the difference in weight the evaporation loss is readily calculated. The amount of asphalt material or tar in the oil left after evaporation is determined, and this gives the amount of carbonization. It is necessary to carry out this evaporation and carbonization test using special apparatus, and many precautions have to be taken to obtain consistent results. The apparatus and method of Waters for determining carbonization are probably most widely used at the present time. The carbon content is the carbon or coke left on distilling the lubricating oil in the absence of air. Conradson's apparatus is usually employed in this country for the determination of the carbon content of oils. The evaporization loss indicates to a certain extent the consumption of oil that will take place in an engine provided the oil is of suitable viscosity.

Carbonization Test

The carbonization test is valuable because it indicates the decomposition that takes place on the cylinder-head and at other places in the engine where the oil is exposed in the form of a thin film to the high temperature. The carbon content of the oil is not the actual amount of carbon present in the oil, but is the amount of coke or carbon left when the oil is heated in the absence of air. The importance of this test is evident when it is realized that the lubricating oil in the cylinder comes into contact with the combustion flame or very hot surfaces and is immediately vaporized, leaving, however, a small amount of carbon behind. The accompanying table gives a summary of the average temperatures for various parts of internal-combustion engines; of course the temperatures vary to a great extent according to the type of engine and the speed at which it is run.

	Temperature, deg. fahr.	
Part	Gas Engine	Gasoline Engine
Maximum temperature of gas	2,730	3,000-3,350
Piston center of face	750	750
Center of exhaust-valve	750	1,300-1,400
Center of inlet-valve	480	...
Cylinder-head	212-480	400-450
Crankcase	...	100-230

The suction stroke in the four-stroke cycle is the only one in which there is a smaller pressure in the combustion-chamber than in the crankcase, and therefore during this stroke leakage of lubricating oil into the cylinder may take place. The extent of this leakage is dependent on the effectiveness of the piston-rings. It is from this oil, drawn past the piston-rings by the suction, that carbon deposits are formed.

Some of the oil will settle on the piston and cylinder-heads and carbonize more or less rapidly according to the nature of the oil; asphaltic material which forms as the oil decomposes giving carbon. There is, of course, present on the cylinder walls an oil-film which is constantly renewed; this film is probably only slightly affected by the heat of combustion, as the temperature of the cylinder wall is comparatively low, and no appreciable carbonization will take place.

However, the major part of the oil getting into the combustion space is probably present as a fine spray. The heat of explosion may be sufficient to burn completely the smaller droplets, but the larger ones will be only partly burned by the momentary high temperature of the explosion. Thus carbon and asphaltic material are formed from the oil. If the carbon so formed is light in texture, it will be blown out of the exhaust in the same way as the carbon produced from the fuel when too rich a mixture of fuel in air is used; if, however, dense asphaltic material is formed, in the ensuing compression stroke it will tend to adhere to the piston-head and cylinder walls, giving rise to carbon deposits. The carbon deposits in an engine, as is well known, do not consist wholly of carbon, but always contain some asphalt and also metallic particles derived from the water in the engine.

Suction Stroke Most Important of the Cycle

According to the views here presented, the suction stroke is the most important of the whole cycle, from the point of view of lubrication; the leakage of oil occurring during this stroke is the origin of the carbon deposits found in engines and is also responsible for the total consumption of the oil in the engine. As it is upon the oil-spray in the combustion space rather than upon the oil-film on the walls that this carbon formation is mainly dependent, the rapid carbonization of oil, that is, the carbon content, will be a more important factor in testing lubricating oils for internal-combustion engines than the gradual carbonization at lower temperatures. The consumption of oil in the engine is also more dependent on the viscosity of the oil used than upon its fire-test.

In lubricating oils there are present small amounts of compounds of a resinous character, and as a result of experimental work it has been shown that the extent of carbonization at high temperatures and the carbon content depend to a great extent on the amount of these resins present. These compounds decompose more readily than the hydrocarbons in the lubricating oils. At the same time it must be recognized that the hydrocarbons in lubricating oil also decompose, giving resinous compounds when exposed to air at high temperatures.

Thus carbon is formed from a lubricating oil as a result of the rapid vaporization of the oil at high temperatures and of the more gradual decomposition of the oil at lower temperatures. These processes take place simultaneously in the internal-combustion engine, and from the results of engine and laboratory tests it appears that the first process, rapid carbonization, is mainly responsible for the formation of carbon deposits and the engine troubles dependent thereon.—F. H. Garner in *Power*.

Standards Committee Meeting

THE meeting of the Standards Committee was convened at 2 p. m. Tuesday May 24, at the West Baden Springs Hotel, West Baden, Ind., with Chairman B. B. Bachman presiding. After declaring a quorum present, Mr. Bachman reviewed briefly the work of the committee during the four-months' period following the annual meeting, after which he called for reports.

The reports of Divisions submitted to and approved in original or amended form by the Standards Committee, the Council and at a session of the Society at West Baden, are given below for letter ballot of the Members of the Society. The letter-ballot forms will be distributed under separate cover and the ballot counted at the Society offices on Saturday July 23. All ballots to be counted must be received on or before that day.

Ball and Roller Bearings Division Report

(1) *Clutch Release Type Thrust Ball Bearings*

Following a request of the Transmission Division that the Ball and Roller Bearings Division establish a separate standard for clutch releases thrust ball bearings, a Subdivision was appointed which formulated a tentative recommendation.

Clutch Release Type Thrust Ball Bearings

Number	Bore	Width	Ball Diameter
1	1 3/8	5/8	5/16
2	1 1/2	11/16	5/16
3	1 5/8	11/16	5/16
4	1 3/4	3/4	5/16
5	1 7/8	3/4	5/16
6	2	3/4	5/16
7	2 1/8	3/4	5/16
8	2 1/4	13/16	3/8
9	2 3/8	13/16	3/8
10	2 1/2	13/16	3/8
11	2 3/4	13/16	3/8
12	3 1/4	7/8	3/8

All dimensions in inches.

These bores and widths are intended for use with bearings with or without assembling bands.

A meeting was held jointly with the Transmission Division on March 14 at which the returns of the general letter were discussed and the proposed recommendation modified. The final recommendation has been approved as given in the accompanying table.

The recommendation does not include the outside diameters of the bearings because of differences in present practice, but the Subdivision plans to give this matter further consideration.

Data available indicated that approximately 80 per cent of the clutch release type bearings now manufactured are in inch sizes, and that a metric standard would not be acceptable.

THE DISCUSSION

W. R. Strickland:—This proposal is the result of several meetings of the Division and Subdivision, and constitutes a boiling down of about 60 sizes of clutch thrust ball bearing now being used. It was found, in considering the various designs of clutch bearing, that it is impossible to standardize on the outside diameters. The Division thought it would be of advantage to standardize the bores, widths and ball sizes in regular inch dimensions. The report is presented as a step in standardization, with the understanding, of course, that in time corrections or additions may be made.

A. E. Brion:—What has been done in the way of cooperating with the British Society?

Mr. Strickland:—No steps have been taken by the Division to cooperate with the British Society. That is not a function of the Division, I believe. Cooperation with organizations in other countries is had through the Sectional Committee on Ball Bearings of the American Engineering Standards Committee.

R. S. Burnett:—This particular series has not been referred to the Sectional Committee as yet. The present standards for ball bearings have, however, been taken up through the American Engineering Standards Committee with European countries with a view to international standardization.

Mr. Brion:—A large part of the proposal we are considering now is what Mr. Renold, of Hans Renold, Ltd., England, suggested some years ago.

M. C. Horine:—If these are light, medium or heavy series bearings, they are very much under the capacity of what is used in some transmissions. Is not the clutch throw-out bearing at least a medium series bearing, and would it not be well to increase the ball diameters?

Mr. Strickland:—This list is selected from all the bearings being used, and the question of capacity was considered. Even the smaller bearings have the necessary capacity for most of the trucks in service today. Diameters are governed more by the individual design of the clutch than the bearing and the size of the balls increases with the bearing diameters. The question of noise affects that, but I think the question of capacity does not enter.

Ernest Wooler:—Is it not usual to express the bore of bearings in decimals? Should the two halves of the bearing not have different sizes, one to fit tightly on the shaft and the other loosely?

Mr. Strickland:—In the present state of this proposed standardization there would be some question about fixing the sizes. The sizes are nominal, fits being prescribed by the various designs.

Chain Division Report

(2) *Roller-Chain Sprockets*

The first tooth-forms for roller-chain sprockets were designed upon the theory that the tooth action is similar to that of gear teeth, the chain being regarded as analogous to a rack. It was, however, soon found that when chains had been in use a short time the wear at the joints caused an elongation of the pitch of the chain, thus destroying perfect registration with the sprocket teeth which resulted in rough action and rapid wear. The sprocket teeth were then redesigned to provide for the chain elongation by making the tooth-gap somewhat greater than the diameter of the chain roller, but with little or no modification in the contour of the tooth, which was so shaped that the chain roller bore almost squarely against the sprocket tooth, throwing practically the whole load on a single tooth.

Meanwhile, the use of roller chains became more extensive and the demands made on them more exacting. This condition naturally led, both in this country and abroad,

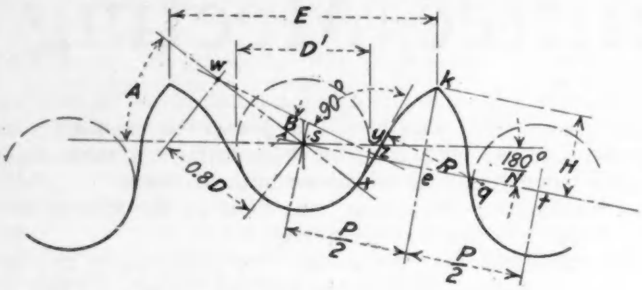


FIG. 1—ROLLER-CHAIN SPROCKET TOOTH-FORM

P — Pitch of chain
D — Nominal roller diameter
N — Number of teeth in sprocket
D' — Diameter of seating curve $= 1.005\ D + 0.003$ in.

$$A = 31° + \frac{90°}{N}$$

$$ws = 0.8\ D$$

$$B = 13° - \frac{18°}{N}$$

xz — A circular arc whose center is at w
zy — A straight line perpendicular to wz

$$xy = 1.3\ D \times \sin\left(25° - \frac{90°}{N}\right)$$

ek — A line perpendicular to st
H — Height of tooth crest above the chord $st = 0.35\ P$
qy — A line parallel to wz with q so located that the circular arc through y will pass through k, thus forming a pointed tooth.

Outside Diameter of Sprocket = Pitch Diameter +

$$P\left(0.7 - \tan\frac{90°}{N}\right) = \text{(approx.) } P.D. + P\left(0.7 - \frac{1.592}{N}\right)$$

$$E = O.D. \times \sin\frac{180°}{N}$$

to a more careful study of the action between chains and sprocket teeth than had theretofore been given to the subject, and it soon became evident that by giving some obliquity to the face of the sprocket tooth, quite different and decidedly better results could be obtained. The action of the forces called into play can be readily analyzed by the graphic method of laying out a parallelogram of forces which will show in true proportion the initial tension on the chain, the resultant thrust on the sprocket tooth (determined by the degree of obliquity selected) and a balancing force passed on to the ensuing links of the chain where this action is repeated in a lessening amount.

By this method the load on a chain can be distributed over several teeth of the sprocket and at the same time, by extending the oblique portion of the sprocket tooth a suitable distance, it is made possible for a chain of elongated pitch to slide outward on this oblique face until it finds a pitch-line on the sprocket coinciding with the lengthened pitch of the chain. These general findings were freely exchanged among the chain manufacturers and one of the English companies published a brochure on the subject. During the war more pressing matters prevented an interchange of opinion on this subject, but nevertheless most of the sprocket manufacturers evolved tooth-forms along these lines.

It is now proposed to effect an agreement upon one standard form that shall supersede the various individual designs and the formula now presented by the Chain Division for adoption as an extension of the present S.A.E. Standard for Roller Chains, page E3, Vol. I, S.A.E. HANDBOOK, is a composite of the best features of the various tooth shapes that have been tried out and pronounced satisfactory.

THE DISCUSSION

MR. HORINE:—This new tooth-form is a proposal to standardize something still in the experimental stage. So far as I know, no trucks have been run for, say, 10,000 miles with this form of chain sprocket. Our company, a

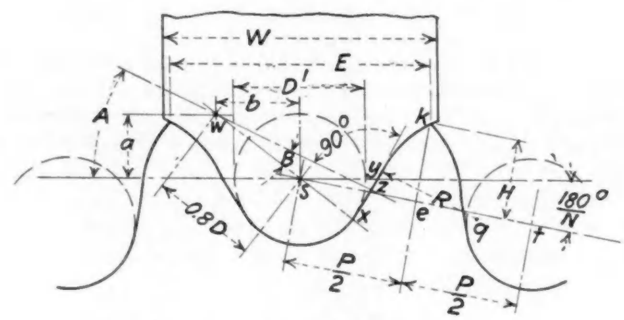


FIG. 2—ROLLER-CHAIN SPROCKET CUTTERS

Cutters shall be designed for 6, 7-8, 9-11, 12-17, 18-34, 35 teeth and over.

The number of teeth on which each cutter shall be based is, $M = \frac{2Nn}{N+n}$, where "N" and "n" are the maximum and minimum number of teeth to be cut by any given cutter. A sprocket having N teeth will thus have a pressure angle which departs the same amount from the desired pressure angle as one having n teeth.

The values of M for the various cutters are respectively, 6, 7.47, 9.90, 14.07, 23.54 and 65.42 teeth.

Cutter numbers shall be respectively, 6, 7½, 10, 14, 24 and 65.

$$D' = 1.005\ D + 0.003 \text{ in.}$$

$$A = 31° + \frac{90°}{M}$$

$$ws = 0.8\ D$$

$$B = 13° - \frac{18°}{M}$$

xz is a circular arc with w as the center
zy is a straight line perpendicular to wz

$$xy = 1.3\ D \sin\left(25° - \frac{90°}{N}\right)$$

$$se = \tfrac{1}{2}\ P$$

$$ek = H = 0.35\ P$$

yq is a line parallel to wz
q = is so located that the arc yk, struck from q as a center, will pass through k

The radius R may be determined graphically.

Where the same roller diameter is used on chains of two different pitches (as ⅝-in. roller, 1-in. pitch; and ⅝-in. roller, 1¼-in. pitch) cutters No. 6 and 7½ shall be designed for the longer pitch, and may be used for cutting sprockets of the shorter pitch. Cutters No. 10, 14, 24 and 65 shall be designed for the shorter pitch and may be used to cut sprockets of the longer pitch.

W = width of cutter $= 1.02 \times O.D. \times \sin\frac{180°}{n}$, to the next higher 32nd inch; where $O.D.$ equals the outside diameter of the sprocket with the larger pitch and the lowest number of teeth to be cut with the cutter.

Note: The specifications to the cutter maker may be simplified if w and z are located as follows:

$$a = 0.8 \times D \times \sin\left(31° + \frac{90°}{M}\right)$$

$$b = 0.8 \times D \times \cos\left(31° + \frac{90°}{M}\right)$$

$$xz = 2.6 \times D \times \sin\left(6\tfrac{1}{2}° - \frac{9°}{M}\right)$$

builder of chain-drive trucks, has not had an opportunity to try out this sprocket. While we do not doubt that it contains great promise, I believe that it is not proper to standardize on a tooth which is still experimental and hypothetical. I think we should let this matter go over for a while until we shall have had an opportunity to try the form out.

H. S. PIERCE:—I would like to say something that will probably give a little clearer view on the matter. The form of cutter that is now in most general use was probably designed by the cutter manufacturers who are more accustomed to working on gears. It is a fact that the chain manufacturers themselves in cutting the wheels have not adhered to that general form of cutter which has been principally supplied by the cutter manufacturers. They learned in the very early days that it was necessary to modify it to get the best results.

We have used since about 1908 cutters and tooth-forms which very closely approximate this present one. One of the elements that led up to this standardization was the fact that the form which is in most general use by others than the chain manufacturers themselves was recognized by them as being entirely unsatisfactory. It is true that this tooth-form has slight modifications from any which is in very general use, but the principles involved have all been thoroughly tried out and the compromises that have been made to have a standard are in no case large enough to influence the result. The proposal is not experimental or indefinite. I am not familiar with the particular type of tooth-form that the company which Mr. Horine represents has been using, but this form has been in very general use for a number of years, and we are trying to put it on a firmer foundation.

CHAIRMAN B. B. BACHMAN:—As I understand it, this tooth-form would not change the design of the chains, that is, the chains as they are being made today would be applicable to the sprockets that Mr. Horine is using as well as to those made by the cutters that we are contemplating adopting.

W. F. COLE:—A standard chain will run on any tooth-form which fits that chain. The question has been asked whether the same chain would run on the old sprocket tooth-form and this new sprocket tooth-form. The answer is yes.

In connection with the remark that this new tooth-form has not been tried out on any truck chain, I can report that the chain and sprocket company which I represent has supplied many sprockets cut in accordance with the principle embodied in this proposal, and that they have been used on trucks which have made thousands and thousands of miles. It works out exceedingly well. I am very sure that the same statement can be made by the other chain and sprocket manufacturers.

MR. BRION:—Mr. Renold suggested this tooth-form in 1913; so it must have been used in England for some time.

MR. PIERCE:—There is one element in standardizing the roller-chain tooth-form that is not of the same character as standardizing other things, in that we do not break interchangeability. The proposal is offered as an improved engineering practice. Any properly designed chain that will run on the old tooth-form will run on this new tooth-form, but it will not run as well on the old one.

(3) *Roller Chains*

The Chain Division's recommendation that the present S.A.E. Standard for Roller Chains, page E3, Vol. I, S.A.E. HANDBOOK, be extended to specify chain numbers in accordance with the following numbering system, was approved.

> The left-hand figures shall denote the number of one-eighth inches in the pitch.
>
> The final figure shall denote the roller diameter as follows: "0", the heavy series roller diameter; "1", the medium series roller diameter; and "2", the light series roller diameter.
>
> The letters "W" or "N" shall denote whether the chains are of the wide or the narrow series.

Thus in Chain No. 160W, 16 indicates that the pitch is 2 in., "0" that it is heavy series roller diameter, and "W" that the chain is of the wide series.

The proper numbers for the present S. A. E. Standard chains of the heavy series are given in the accompanying table.

PROPOSED NUMBERS FOR S. A. E. STANDARD HEAVY SERIES ROLLER CHAINS

Chain Number	Pitch	Roller Diameter	Chain Width
30W	3/8	0.2500	0.2500
40W	1/2	0.3125	0.3125
50W	5/8	0.4000	0.3750
60W	3/4	0.4690	0.5000
80W	1	0.6250	0.6250
100W	1 1/4	0.7500	0.7500
120W	1 1/2	0.8750	1.0000
140W	1 3/4	1.0000	1.1250
160W	2	1.1250	1.2500
200W	2 1/2	1.5500	1.5625
240W	3	1.9000	1.8750
320W	4	2.5000	2.5000
400W	5	3.0000	3.0000

All dimensions in inches.

ELECTRICAL EQUIPMENT DIVISION REPORT

(4) *Insulated Cable*

The Electrical Equipment Division's recommendation that the present S.A.E. Standard for Insulated Cable, page B33, Vol. I, S.A.E. HANDBOOK, be revised to conform to the accompanying specifications for insulated cable, was approved. Although the proposal does not as yet include electrical tests, it is desired by the cable manufacturers that the manufacturing specifications be adopted and published without delay so far as they have been completed.

The electrical tests for high-tension ignition cable will be included in the standard at a later date, as no entirely satisfactory tests have yet been developed. The electrical tests included in the present standard, although used considerably in general power-cable testing, are not considered satisfactory for automotive high-tension ignition cable.

The proposed revision is submitted as a practical and satisfactory specification founded on the best commercial experience. It embodies many points in common with Government specifications which have been generally approved and will be used by manufacturers in producing high-grade insulated cable.

INSULATED CABLE

I. General Specifications

Conductors.—Conductors shall be bunched or stranded as specified in each section and shall be of annealed copper wire in accordance with Specification No. B3-15 of the American Society for Testing Materials. All wires shall be thoroughly tinned and must withstand the tinning test as specified in Section II Tests. All tests of copper conductors shall be made before stranding or insulating.

Cotton Separators.—Material for separators, where specified, shall be of good grade cotton and shall be closely and tightly applied.

Rubber Insulation.—Rubber insulations shall be homogeneous in character, properly vulcanized, and placed concentrically about the conductors.

Rubber insulations shall adhere closely to, but shall strip readily from, the conductors, leaving them reasonably clean.

Rubber insulations used on cables covered by these specifications shall contain not less than 20 per cent (by weight) of good grade Hevea rubber which has not been previously used.

Varnished Cambric Tape.—Varnished cambric tape shall be made from a good grade cotton fibre treated with multiple coats of insulating varnish. The instantaneous puncture voltage shall be not less than 750 volts per mil of thickness tested in accordance with the standards of the American Institute of Electrical Engineers.

Varnished cambric tape shall be not less than 0.005 in, nor more than 0.013 in. thick.

Braids.—Braids shall consist of closely woven cotton yarn, and shall not be less than 1/64 in. thick. Braids shall be impregnated with at least two coats of properly dried, heat, oil and water resisting insulating varnish or impregnated with black weather-proof compound which has an even and smooth finish. Adjacent layers of cable, when wound on the reel, shall not stick to one another at any temperature under 105 deg. fahr. (40 deg. cent.).

Armor.—Armor shall be of either galvanized or sherardized soft steel, soft brass, aluminum or copper and applied in a close helix. Successive turns shall not overlap. Armor dimensions shall be as given in Table 1.

Armor shall be solid "D" shaped, unless otherwise specified by the purchaser.

The large armor is recommended for use on all cables exceeding ½-in. diameter underneath the armor.

TABLE 1—ARMOR THICKNESS AND WIDTH DIMENSIONS

Armor	Thickness, In.			Width, In.		
	Min.	Nom.	Max.	Min.	Nom.	Max.
Small	0.014	0.017	0.020	0.045	0.050	0.055
Large	0.017	0.020	0.023	0.095	0.100	0.105

II. Tests

Tinning Test.—For this test, samples of the bare wire before being stranded or insulated shall be properly selected to secure an average grade of tinning. The wires shall be thoroughly cleansed by means of ether, benzine, gasoline, naphtha, caustic alkali solution, alcohol, or hot water and soap, whichever may be found necessary to thoroughly clean the wires.

The wires shall then be rinsed in clear water and wiped dry with a soft cotton cloth. The wires shall then be immersed for 1 min. in a solution of hydrochloric acid having a specific gravity of 1.088 at 70 deg. fahr. (21 deg. cent.), and then rinsed in clear water and wiped dry as above specified. The wires shall then be immersed for 30 sec. in a solution of sodium polysulphide which contains an excess of sulphur and which has sufficient strength to thoroughly blacken a piece of clean untinned copper wire in 5 sec.

The complete cycle of operations shall then be repeated, commencing with the immersion in hydrochloric acid and ending with the immersion in the sodium polysulphide solution.

Tests of tinning shall be made on not less than 10 sets of samples of reasonable length. All wires shall withstand one immersion in the hydrochloric acid without blackening in the sodium polysulphide solution, and 75 per cent of the wires shall withstand three immersions in the hydrochloric acid without blackening in the sodium polysulphide solution. All tests shall be conducted with the solutions at a temperature of 70 deg. fahr.

Physical Tests.—A test-specimen of rubber insulation, which has not previously been handled, not less than 6 in. long shall have marks placed upon it 2 in. apart. The sample shall then be stretched at the rate of 12 in. per min. until these marks are 6 in. apart, and then immediately released. Thirty seconds after being released the distance between the marks shall not exceed 2½ in. The test-specimen shall then be stretched until the marks are 7 in. apart before it is ruptured.

The ultimate tensile strength of rubber insulation shall not be less than 600 lb. per sq. in. The tensile strength shall be calculated upon the original cross-section of the test-specimen before stretching.

Physical tests shall be made at a temperature of not less than 50 deg. fahr. (10 deg. cent.), nor more than 90 deg. fahr. (32 deg. cent.).

For the purpose of these tests, care must be used in cutting to obtain samples of uniform cross-section and no manufacturer shall be responsible for results obtained from samples imperfectly cut.

The above physical tests shall not apply to wires or cables having a wall thickness of less than 0.045 in. For wires and cables having a wall thickness of less than 0.045 in. the initial and ultimate stretch shall be 5 and 6 in. respectively, and the tensile strength not less than 500 lb. per sq. in.

Miscellaneous Tests.—The following tests apply to high-tension (secondary) ignition cables only.

Oil Test for Braided Cables.—A sample of cable shall be immersed in a mixture of equal parts of machine oil and gasoline for a period of 24 hr. without allowing the ends of the sample to become submerged. After this immersion the impregnating varnish should not show signs of softening or absorption, and when the braids have been peeled off, it should be shown that no oil has penetrated to the rubber insulation.

III. Specifications for High-Tension (Secondary) Ignition Cables

Conductors shall be stranded and covered with rubber insulation.

High-tension (secondary) ignition cables shall be

TABLE 2—HIGH-TENSION (SECONDARY) IGNITION CABLE SIZES

Nominal Size		Number of Wires in Strand	Nominal Size of Wires in Strand, A.w.g.	Maximum Outside Diameter, In.	Minimum Outside Diameter, In.	Minimum Thickness of Rubber Wall, In. (Plain Rubber Covered)	Minimum Thickness of Rubber Wall, In. (Single Braid)	Minimum Thickness of Rubber Wall, In. (Double Braid)
Mm	In.							
7	0.2756	12 19	26 (0.0159) 27 (0.0142)	0.285	0.265	0.097	0.081	0.066
9	0.354	19 19	27 (0.0142) (0.0147)	0.364	0.344	0.135	0.119	0.104

The 7-mm. size is recommended for all high-tension cable.

plain rubber covered, single braided, rubber face taped and single braided or double braided. Weatherproof braid shall not be used on this type of cable.

High-tension (secondary) ignition cable sizes shall be as shown in Table 2.

IV. Specifications for Low-Tension (Primary) Ignition Cables

Conductors shall be bunched or stranded and covered with rubber insulation.

Low-tension (primary) ignition cable shall be plain rubber covered, single braided, rubber face taped and single braided, or double braided.

Low-tension (primary) ignition cable sizes shall be as shown in Table 3.

TABLE 3—LOW-TENSION IGNITION CABLE SIZES

Nominal Size	Number of Wires in Strand	Nominal Size of Wires in Strand		Maximum Outside Diameter, In.	Minimum Outside Diameter, In.
		A.w.g.	In.		
5 mm. (0.197 in.)	12 19	26 27	0.0159 0.0142	0.207	0.187

V. Specifications for Rubber Covered Lighting and Starting Cables

Conductors of cables Nos. 16 to 10 A.w.g. inclusive shall be either bunched or stranded as desired. Stranded construction is recommended for flexibility. Conductors of cables No. 8 and larger shall be stranded and may be either concentric or rope lay.

Conductors shall be covered with rubber insulation.

Note: Lighting and starting cables shall be single braided, rubber face taped and single braided, or double braided.

Lighting and starting cable sizes shall be as shown in Table 4.

TABLE 4—STRANDING AND DIMENSIONS OF LIGHTING AND STARTING CABLE

Nominal Size, A.w.g.	Number of Wires in Strand	Nominal Size of Wires in Strand		Circular Mils		Continuous Carrying Capacity, Amp.	Maximum Outside Diameter, In.	Minimum Thickness of Rubber Wall, In.
		A.w.g.	In.	Nominal	Actual			
	12	27	0.0142		2,418			
16	16	28	0.0126	2,583	2,557	6	0.200	0.022
	19	29	0.0112		2,407			
	19	27	0.0142		3,829			
14	26	28	0.0126	4,107	4,155	15	0.223	0.027
	19	25	0.0179		6,088			
12	26	26	0.0159	6,530	6,607	20	0.250	0.031
	19	23	0.0225		9,681			
10	49	27	0.0142	10,383	9,873	25	0.275	0.031
	19	21	0.0284		15,392			
8	51	25	0.0179	16,510	15,680	35	0.320	0.037
4	61	22	0.0253	41,741	39,193	70	0.420	0.0468
	61	20	0.0319		62,312			
2	127	23	0.0225	66,371	64,707	90	0.490	0.0468
	127	22	0.0253		81,598			
1	133	22	0.0253	83,693	85,453	100	0.600	0.0625
	127	21	0.0284		102,883			
0	133	21	0.0284	105,535	107,743	125	0.635	0.0625
	127	20	0.0319		129,731			
00	133	20	0.0319	133,077	133,817	150	0.700	0.0625
	259	23	0.0225		131,961			

VI. Specifications for Varnished Cambric Insulation Lighting and Starting Cables

Conductors shall be constructed as described in Section V, and shall be stranded as shown in Table 4.

Lighting and starting cables of this class shall have two or more layers of overlapping varnished cambric tape. Alternate layers shall be laid in opposite directions.

Lighting and starting cables of this class may be either single or double braided.

TABLE 5—ADDITIONAL SPECIFICATIONS FOR VARNISHED CAMBRIC INSULATION AND ARMORED LIGHTING AND STARTING CABLES

Nominal Size, A.w.g.	Continuous Carrying Capacity, Amp.	Maximum Outside Diameter Varnished Cambric Cables, In.	Maximum Outside Diam. Armored Cables, In.
16	8	0.215	0.255
14	18	0.229	0.269
12	22	0.249	0.289
10	27	0.273	0.313
8	45	0.298	0.338
4	80	0.383	0.423
2	110	0.450	0.496
1	140	0.530	0.576
0	180	0.570	0.616
00	210	0.629	0.675

VII. Specifications for Armored Lighting and Starting Cables

Conductors shall be constructed as described in Section V and shall be stranded as shown in Table 4.

Lighting and starting cables of this class shall have two or more layers of overlapping varnished cambric tape. Alternate layers shall be laid in opposite directions.

Lighting and starting cables of this class may be either single or double braided.

THE DISCUSSION

A. D. T. LIBBY:—It has been the task of the Subdivision that prepared this proposal to develop a new specification more in line with present automotive practice. The proposal does not include any electrical tests as the Subdivision does not feel that satisfactory electrical tests have been devised for automotive high-tension insulated cable. Many types of rubber-insulated wire and cable now on the market can be subjected to a high-voltage test at, for instance, 60 cycles, and some will break down at 20,000 volts, while some will withstand as much as 25,000 volts. Yet, when such cable is tested under certain conditions with the high frequency of the ignition system, the cable that shows the higher break-down voltage when subjected to the regular test at 60 cycles, will fail much more

quickly than a cable which would break down at a lower voltage.

W. S. HAGGOTT:—I have a letter from Mr. Burley, one of the members of the Subdivision which prepared this report, referring to the note under table 2, stating that the 7-mm. size is recommended for all high-tension cable. Mr. Burley writes:

> I do not agree with this recommendation. In the first place, it seems to me that it is not the function of these specifications to advise which size of high-tension cable to use, and, secondly, I think the 9-mm. size is more reliable than 7-mm. about in proportion to the difference in cost. The 9-mm. is called for by several of the best informed electrical engineers connected with the automobile plants. Nearly one-half of the plain rubber high-tension cable we make is in the 9-mm. size. I wish to dissent from the recommendation.

Although we do not find that the 9-mm. size is 50 per cent of our high-tension cable production, a large quantity of it is used, and I see no objection to deleting the reference note in the proposal.

MR. LIBBY:—Although the 9-mm. cable is used in the manufacture of ignition systems, the greater portion of it seems to be 7 mm. and judging from all the tests that I have made in the laboratory and elsewhere, the 7-mm. cable seems to be perfectly satisfactory. It is the size that has become standard on motorcycles, on which the cables are subjected to mud, oil, sunlight and water and have to stand up under much more severe service than on an automobile. In view of the fact that the large bulk of the cables seems to be of the 7-mm. dimension, it would seem advisable to retain the footnote.

H. T. WREAKS:—Our experience with the 9-mm. size was one of development due to the 7-mm. cable as made having failed to meet conditions on certain prominent makes of automobile on which the duty happened to be particularly heavy. It seems to me that simply to make this recommendation does not help the specifications in any way, and to do so would reflect more or less upon the practical experience of the manufacturers who have found a need for the 9-mm. cable and made all their dies and fittings for it.

W. A. CHRYST:—I believe the specification should favor the 7-mm. cable not so much on account of the changes in the size of wire as of price and to keep the number of sizes of terminals and fittings which go on the electrical equipment at a minimum.

At the present time ignition manufacturers have to make dies for 9-mm. as well as 7-mm. cable. There is nothing in the recommendation which prevents using the 9-mm. size, but where the matter is open there is no hesitancy in recommending the use of the 7-mm. size.

I can confirm Mr. Libby's remarks. In the case of motorcycles and airplanes, where the service is apparently very much more severe than on automobiles, no criticism has ever been brought against the 7-mm. cable.

MR. WREAKS:—Is there any reason for specifying that where lighting and starting cables are specified, they are limited to varnished cambric tape? Much rubber-covered armored cable is made and used.

MR. HAGGOTT:—I am of the opinion, which I think is concurred in by other cable manufacturers, that it is possible to make a good rubber-covered armored cable which will meet all of the possible requirements of starting and lighting service, but I think that the rubber-covered armored cable construction has not been developed to the point where it ought to be considered for a standard. Sometime an armored rubber-covered cable will be developed that will probably be as satisfactory as the present varnished cambric construction.

MR. STRICKLAND:—I notice that it is specified in the recommendation that high-tension cable should be rubber-covered. No provision is made for composition coverings, of which there are possibly several that are just as good as or better than rubber.

MR. LIBBY:—That is covered in the proposal, under

> High-tension ignition cables shall be plain rubber covered, single braided, rubber face taped and single braided or double braided.

(5) *Electrical Equipment Nomenclature*

Last fall it was suggested that the S. A. E. Standard Automobile Nomenclature for electrical equipment be revised to include terms applying to equipment which have been developed and come into general use since the present standard was adopted. The Division appointed a Subdivision to review this part of the standard nomenclature and to prepare a report which would bring it up-to-date. This report has been submitted to and reviewed by the Electrical Equipment Division and the recommendation approved for adoption by the Society. The report refers to the Divisions and Groups indicated, which are printed on pages K8 and K9, Vol. I, S. A. E. HANDBOOK.

DIVISION VII—IGNITION

Group 1—(Same as present standard)

Group 2—Battery Ignition Equipment

- Ignition Set
- Ignition Coil
- Ignition Switch
- Timer-Distributor
- Breaker-Arm
- Breaker Contacts
- Breaker Cam
- Distributor Rotor
- Distributor-Rotor Brush
- Distributor Cap
- Timer-Distributor Shaft
- Timer-Distributor-Shaft Gear
- Ignition Drive-Shaft
- Ignition Drive-Shaft Gear
- Manual-Advance Arm
- Automatic-Advance Element
- Ignition Unit, Magneto-Base Mounting

Group 3—Magneto (Same as present standard)

Group 4—(Omitted)

DIVISION VIII—STARTING AND GENERATING EQUIPMENT

General

- A single-unit system comprises a starter-generator
- A separate-unit system comprises a generator and a starting motor separately mounted
- A combined-unit system comprises a duplex starter-generator, an ignition-generator, or an ignition-starter-generator
- Direction of rotation is "clockwise" or "counter-clockwise" as determined by the driven shaft for magnetos, generators, starter-generators, and by the driving shaft for starting motors
- Methods of mounting units are: flange, base, strap and sleeve

Group 1—Generator

- Generator
- Generator Main Brush
- Generator Main Brush-Holder
- Generator Third Brush
- Generator Third Brush-Holder
- Generator Field Frame

Generator Field Fuse
Generator Driving Gear or Sprocket
Generator Shaft
Generator Coupling (Members as indicated under Magneto Coupling)

Group 2—Starting Motor

Starting Motor
Starting-Motor Brush
Starting-Motor Brush-Holder
Starting-Motor Pinion
Starting-Motor Intermediate Gear
Starting-Motor Intermediate-Gear Shaft
Starting-Motor Intermediate Pinion
Manual Shift
Screw Shift
Magnetic Shift

Group 3—Starter-Generator (Parts covered by Division VIII, Groups 1 and 2. Group 3 in the present standard is omitted.)

Group 4—Ignition-Generator (Parts covered by Division VII, Group 2, and Division VIII, Group 1.)

Group 5—Ignition-Starter-Generator (Parts covered by Division VII, Group 2, and Division VIII, Groups 1 and 2.)

Group 6—Battery (Same as Group 4 in present standard.)

DIVISION IX—MISCELLANEOUS ELECTRICAL EQUIPMENT

Group 1—(Same as the present standard except as revised by the Lighting Division.)

Group 2—Switches and Instruments

Lighting Switch
Starting Switch
Ignition Switch
Combined Switch (Such as lighting-ignition)
Ammeter
Voltmeter
Volt-Ammeter
Charging Indicator
Cut-Out Relay
Cut-Out Relay Contacts
Cut-Out Relay Armature
Cut-Out Relay Shunt Coil
Cut-Out Relay Series Coil
Current Regulator
Voltage Regulator
Starting-Switch Contacts
Starting-Switch Contactor
Starting-Switch Plunger (Lever or Button)
Through-the-Board Mounting
Front-of-Board Mounting
Ground-Return Wiring
Insulated-Return Wiring

Group 3—Horn

Motor-Operated Horn
Vibrator Horn
Hand Horn
Horn Projector
Horn Diaphragm
Horn Sound Ratchet
Horn Motor

Group 4—Miscellaneous (Same as the present standard)

THE DISCUSSION

MR. STRICKLAND: — In the proposed nomenclature timer-distributor might better be called ignition-distributor, which is a term more generally used.

MR. LIBBY:—The Division discussed that point and finally agreed that timer-distributor is the better term to use, because the timer is built into the distributor in the majority of cases. That is to say, there is the distributor head, underneath the head is the distributor finger, and usually below that is the timer which determines the time at which the spark takes place.

MR. HORINE:—Why call it a timer on a battery system and a breaker on a magneto system?

MR. LIBBY:—This is largely a matter of usage. In connection with a distributor it is called a timer-distributor, but there is not exactly the same arrangement on a magneto, where the part has always been referred to as a breaker, never as a timer.

MR. HORINE:—In the old battery systems there was a timer which looked very much like a distributor. There is some danger of confusion in this connection.

MR. STRICKLAND:—The term timer-distributor has not been employed at all by users, I think. Possibly it has been used by magneto manufacturers.

MR. CHRYST:—I think that none of the manufacturers uses as a whole the terms recommended. This proposal was the best compromise that would be intelligible to all the manufacturers. It is customary for some to call the complete unit a distributor, while others call it a battery system, but to describe this apparatus definitely, without being too verbose, the name timer-distributor seemed to fit as well as anything.

C. E. WILSON:—We call it either a timer or a distributor but do not use the two terms together.

MR. CHRYST:—That is what we found among the members of the Division, and it seemed that timer-distributor was the best compromise, because it describes the apparatus accurately and is something we could all agree on.

H. E. CLAY:—I notice that the report refers to timer-distributor and then in naming the related parts mentions breaker-arm, breaker contacts, breaker cam, distributor rotor, distributor-rotor brush and distributor cap. Would not breaker-distributor do?

MR. LIBBY:—There seems to be some misunderstanding regarding this term. As Mr. Chryst has pointed out, this apparatus is known in different factories by various names, and the proposed term is more or less of a compromise. A timer-distributor is the complete unit, comprising the breaker-arm, breaker contacts, breaker cam, distributor rotor, distributor-rotor brush, distributor cap, the shaft, and the whole assembly ready to install in an automobile.

MR. STRICKLAND:—The term distributor being used, to be consistent the word timer should be eliminated and the head called a distributor.

MR. LIBBY:—The unit is either an ignition distributor or a timer-distributor.

W. M. BRITTON:—The definition as recommended by the Division tells the whole story; it is a timer and distributor combined. The word distributor would designate a distributor alone without the combination of a breaker or timer of any description. In other words, in a magneto there is a distributor as one unit and also a breaker-box containing a breaker. It is not a combination unit, as called for and designated by this nomenclature. I believe the nomenclature should stand exactly as recommended.

A. J. SCAIFE:—I believe it is superfluous to call it that; it sounds like "free gratis." When you refer to distributor, you refer to the whole instrument, and there is no other part of the car that it can be confused with. There is nothing on the car which has anything like that same name, and when talking about a distributor, you mean a distributor or timing device.

G. W. DUNHAM:—A few years ago, when we had air starters, we had an air distributor. We do not know that we shall not have other kinds of distributor. At the present time, I think, there are oil distributors. Distributor is a broad term which means nothing unless

there is some other word with it to explain what kind of a distributor is meant.

MR. BRITTON:—One type of magneto now being built has a distributor similar to that used on battery systems so far as the distributor is concerned. If this nomenclature is changed to distributor, it would designate that type of unit. That is not what is meant at all. The reference is to a combination unit, a timer for the primary system and a distributor for the secondary system.

P. M. HELDT:—Instead of the word timer, I think we should use breaker. Timer is an old word and is not used any more.

CHAIRMAN BACHMAN:—You have a Division which has worked on this subject. Nine of them have voted "yes"; none of them has voted "no"; two of them have not voted. These men are well acquainted with this situation, have sat together in a small meeting, and have thought this thing out. You have their recommendation. There is no doubt that there is a reason for a division of opinion on the matter, and probably the ideas you have expressed here are as good as those your Division has reported. The thing to do is to vote according to what you think the term should be.

(6) *Magneto Couplings*

The present standard for magneto flexible-disc couplings printed on page B14, Vol. I, S. A. E. HANDBOOK, was criticised at the January Standards Committee Meeting as being too distinctive of but one type of coupling, and was referred back to the Electrical Equipment Division for further consideration. The Division has investigated the various types of coupling in more or less common use, and recommended that the present standard for magneto flexible-disc coupling be cancelled, and that the following note be substituted under the drawing of the S. A. E. Standard Magneto Mounting on page B14

> Magneto couplings shall be 2⅜ in. long and have an outside diameter not greater than 3¼ in.

The Electrical Equipment Division has recommended to the Parts and Fittings Division that a standard be formulated for a complete line of flexible-disc couplings.

ENGINE DIVISION REPORT

(7) *Carbureter Flanges—Cast-Iron Type*

The Engine Division's recommendation that the present S.A.E. Recommended Practice for cast-iron carbureter flanges be cancelled has been approved. The present recommended practice reads:

> The flange dimensions for each nominal size cast-iron or oversize carbureter shall be the same as the flange dimensions for the next larger size two or four-bolt carbureter given on pages A8 and A9.

This recommended practice was formulated by the Stationary Engine and Lighting Plant Division in 1919 at which time cast-iron carbureters were coming into use for stationary and tractor engines.

(8) *Carbureter Flanges—Two-Bolt Type*

Experience has indicated that cast-iron carbureter and fitting flanges made to the dimensions given in the S.A.E. Recommended Practice printed on page A8, Vol. I, S.A.E. HANDBOOK, but with the flanges somewhat thicker than for brass, are satisfactory for all general purposes. Such practice makes unnecessary the oversizing of flanges for cast iron. The following recommendation of the Engine Division was approved for inclusion in the present recommended practice.

> When carbureter and fitting flanges are made of brass, their dimensions shall conform to the accompanying table. When they are made of cast iron, the flange thickness, dimensions *F* and *G*, shall be increased ⅛ in. for all sizes.

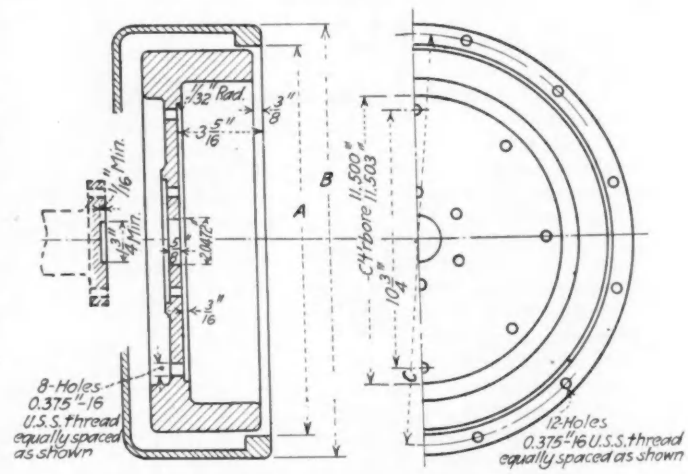
(9) *Disc-Clutch Flywheel Housings*

The Engine Division received a request from one of the transmission manufacturers to include a dimension for the thickness of the flywheel web where it is bolted to the crankshaft of the engine, inasmuch as considerable inconvenience has been caused with regard to fitting the pilot bearing into the present standard bore of 2.0472 in.

Data were collected of dimensions used on the principal makes of engine, and the Division's recommendation approved to add the following dimensions to the present S.A.E. Standard for Disc-Clutch Flywheel Housings printed on page A1, Vol. I, S.A.E. HANDBOOK.

> Crankshaft flange counterbore: minimum diameter, 1¾ in.; minimum depth, 1/16 in.
> Flywheel web thickness of counterbore: ⅝ in.
> Clutch counterbore in flywheel: diameter limits, 11.500 and 11.503 in.; depth, 3/16 in.; corner radius, 1/32 in.

The dimensions for crankshaft flange bolt clearance space have been added to the drawing and accord with the dimensions adopted at the Annual Meeting of the Society.



Size No.	A	B	C
1	20⅛	21¾	20⅞
2	17⅝	19¼	18⅜
3	16⅛	17¾	16⅞
4	14¼	15⅞	15
5	12⅜	14	13⅛

The minimum diameter of the clearance space for crankshaft flywheel bolt-heads shall be 6⅛ in. and the minimum depth ⅝ in. All dimensions in inches. Fixed dimensions do not apply to size No. 5.

THE DISCUSSION

S. O. WHITE:—The report specifies a tolerance of 0.003 in. on an 11½-in. diameter counterbore. Would there be any objection to changing that tolerance from plus 0.003 in. to plus 0.002 in.?

J. B. FISHER:—The tolerance of 0.003 in. is necessary on account of the large diameter. It is much easier to hold to a 0.003-in. limit on a piece that is ground to size than on a large flywheel disc machined on the Bullard or

other heavy-duty machines of that type. Three-thousandths of an inch is about the minimum tolerance that the engine builders can work to in such places.

T. C. MENGES:—How much clearance will that counterbore give for the outside of the bolt-heads?

MR. FISHER:—One-sixteenth of an inch at all points, on the sides and at the end.

ISOLATED ELECTRIC LIGHTING PLANT DIVISION REPORT

(10) *Rating of Storage Batteries*

In August, 1919, the Society adopted the present standard printed on page B37, Vol. I, S.A.E. HANDBOOK, as a result of the proposal formulated by a Subdivision of the Electrical Equipment Division and approved by the Electrical Equipment Division. The present standard was approved by the isolated electric lighting plant manufacturers as represented by that group in the Gas Engine and Farm Power Association (then the National Gas Engine Association). About one year later this group of the Gas Engine and Farm Power Association held a meeting at which the misunderstandings and confusion which had arisen between the manufacturers and their customers, were discussed. The majority of lighting plant manufacturers who participated felt that the standard was unsatisfactory, and formally requested the Isolated Electric Lighting Plant Division to recommend the cancellation of the present standard and to formulate a new one which would be simpler and suitable for adoption by all the lighting plant manufacturers. The Division has given this subject careful consideration and has held several meetings at which non-members of the Division representing the lighting plant and storage battery manufacturers have been present. Early this year a meeting was held at which it was agreed that the lighting plant manufacturers would consider some rating for storage batteries expressed in terms of watt-hours or kilowatt-hours in order that the rating might be expressed in the same terms as the ratings used for electrical equipment operated on isolated electric lighting plants, such as vacuum cleaners, washing machines, sad irons and electric incandescent lamps. At a subsequent meeting other suggested ratings were carefully considered, among which was one expressed in terms of the hours of discharge of the battery when discharged at a constant rate corresponding to 300 watts or fifteen 20-watt lamps, based on a nominal voltage of 2 volts per cell.

It was understood that this new method of rating could be readily adopted by the lighting plant manufacturers who are at present using different methods, such as the 72-hour intermittent and the constant 8-hour discharge ratings. It was thought very desirable to have a common rating which could be universally adopted and conformed to by the lighting plant manufacturers. It was realized that the proposed new rating would be used very largely as a commercial one, although it would enable the user of the plants to make reliable comparative tests on the batteries he buys.

The recommendation of the Division was referred to a meeting of the Farm Lighting Plant Group of the Gas Engine and Farm Power Association held in Chicago April 28 with the intention of securing unanimous approval of the Division's proposal by all the lighting plant manufacturers. It was approved in general at this meeting except that the discharge rate was reduced to correspond to 200 watts or ten 20-watt lamps instead of 300 watts. The recommendation of the Division was therefore printed in two sections and each considered separately by the Standards Committee and Society.

[Section 1 of the recommendation as printed below was approved. Section 2, which referred to the adoption of a new rating, was referred back to the Division. This section of the recommendation together with the discussion at the Standards Committee meeting is printed on page 70 of this issue of THE JOURNAL under the recommendations which were not accepted by the Standards Committee.]

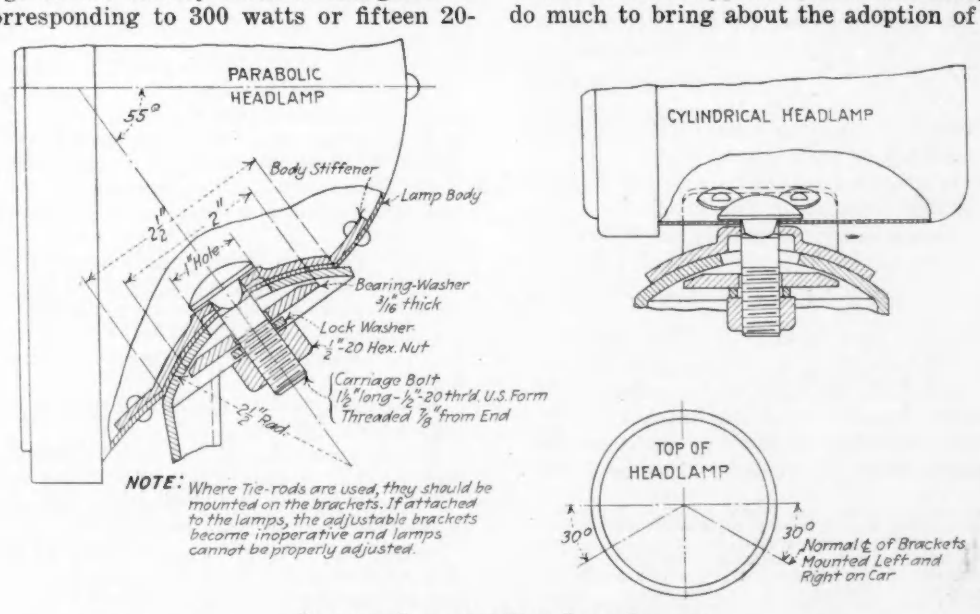
SECTION 1

The Isolated Electric Lighting Plant Division recommends that the present S. A. E. Standard "Rating of Storage Batteries" printed on page B37, Vol. I, S.A.E. HANDBOOK, be cancelled.

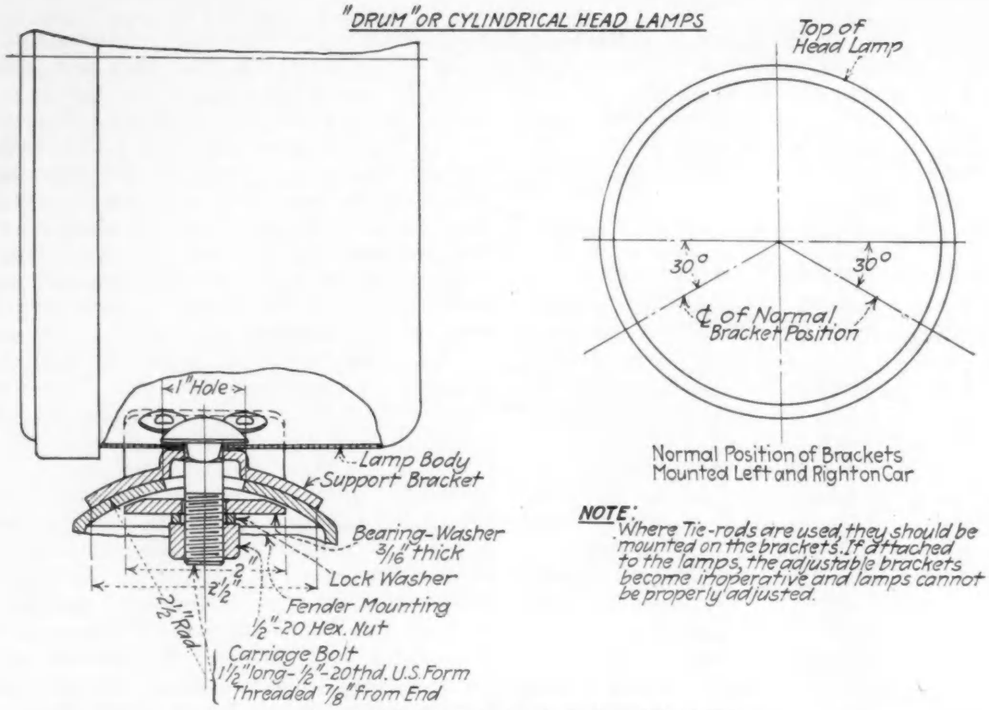
LIGHTING DIVISION REPORT

(11) *Head-Lamp Brackets*

In August, 1920, the Society adopted a recommendation formulated by the Lighting Division for a fender type of head-lamp bracket. This recommendation applied to the bolt-hole dimensions for attaching the bracket to the fender and was approved, as it was thought that it would do much to bring about the adoption of a universal type



PROPOSED PARABOLIC HEAD-LAMP BRACKET
Brackets shall be fastened to the fenders by two ⅜-in. bolts set 3 in. apart center to center



PROPOSED CYLINDRICAL HEAD-LAMP BRACKET
Brackets shall be fastened to the fenders by two ⅜-in. bolts set 3 in. apart center to center.

of mounting on which the Lighting Division was working at that time.

As a result of a large amount of work by the Lighting Division and the lamp manufacturers in cooperation with a number of automobile engineers, a universal fender-type head-lamp bracket has been developed and found satisfactory in actual practice. It will permit the universal adjustment of the lamp and interchangeability of lamps.

The Lighting Division's recommendation will supersede the present recommended practice for the fender-type head-lamp mounting, page B1, Vol. I, S.A.E. HANDBOOK.

THE DISCUSSION

MR. WOOLER:—What is to prevent the bolt from backing out and damaging the reflector?

C. E. GODLEY:—A piece riveted on the inside keeps the bolt from coming out. This is not shown on the drawing in the report as such details are left to the lamp manufacturers.

MR. WOOLER:—How will all lamp manufacturers know about it unless it is shown or mentioned?

MR. GODLEY:—This proposal refers only to the adjustable and interchangeable features of the bracket.

MR. CHRYST:—Is this a patented arrangement?

MR. GODLEY:—No.

(12) *Motorcycle Head-Lamp Mounting*

The Lighting Division's recommendation that the present S.A.E. Standard for Motorcycle Head-Lamp Mounting, page B2, Vol. I, S.A.E. HANDBOOK, be extended to specify 5½ in. between the centers of the bracket prongs was approved. This dimension is used by most of the motorcycle manufacturers at the present time and will make the existing standard much more effective in interchangeability of motorcycle head-lamps.

(13) *Lamp Nomenclature*

As the many variations in the manner of specifying types of lamps has caused a good deal of misunderstanding and trouble for lamp manufacturers and users, the standardization of lamp nomenclature in respect to type, position and use has been worked out by the Lighting Division. It was felt that the nomenclature should indicate the several general types of lamps only, as there are several variations of different types which are usually fully described in the purchase specifications, the purpose of the proposal being to establish common terms which will be used by everybody.

Head-Lamp.[1]—A lighting unit on the front of a vehicle intended primarily to illuminate the road ahead of the vehicle.

Side-Lamps.[2]—A lighting unit mounted on either side of a vehicle and intended primarily as a marker to indicate the location of the vehicle.

Tail-Lamp.—A lighting unit used to indicate the rear end of a vehicle by means of a ruby light.

Backing-Lamp.—A lighting unit mounted on the rear end of a vehicle and intended to illuminate the road to the rear.

Spot-Lamp.—A lighting unit mounted on a manually operated adjustable bracket, which has one focusing type reflector and one focusing type light source.

Instrument-Lamp.—A lighting unit mounted on the instrument-board and intended to illuminate the instruments.

Dome-Lamp.—An interior lighting unit mounted in the top of a vehicle.

Panel-Lamp.—A lighting unit mounted either in the rear panel or in the corners of a closed vehicle.

Tonneau-Lamp.—A lighting unit mounted in the back of the front seat in open or closed vehicles.

Step-Lamp.—A lighting unit mounted on the exterior of a vehicle and intended primarily to illuminate the step or running-board.

Hood-Lamp.—A lighting unit mounted under the

[1] Head-lamps may be divided into three general types: single-socket, a lamp having one focusing type reflector and one focusing type light source; two-socket, a lamp having one focusing type reflector and one focusing type and one auxiliary light source; and duplex, a lamp having two focusing type reflectors and two focusing type light sources.

[2] Side-lamps cover such types as are generally known as bullet, cowl, fender or parking, pillar or windshield lamps.

hood of a vehicle to illuminate the engine compartment.

Inspection-Lamp.—A portable lighting unit connected by an extension cord to the lighting system of a vehicle.

(14) *Bases, Sockets and Connectors*

The Subdivision on Bases, Sockets and Connectors formulated revisions of the present S.A.E. Standard for Bases, Sockets and Connectors, pages B4 and B5, Vol. I, S.A.E. HANDBOOK, which the Lighting Division members have carefully reviewed and consider will bring the present standard into accord with the best engineering practice. The recommendation of the Division was approved and includes the following changes:

Bases.—Specify a dimension of 0.040 in. as a maximum height of solder on the finished lamp to prevent having a heavy slug of solder on the lamp base.

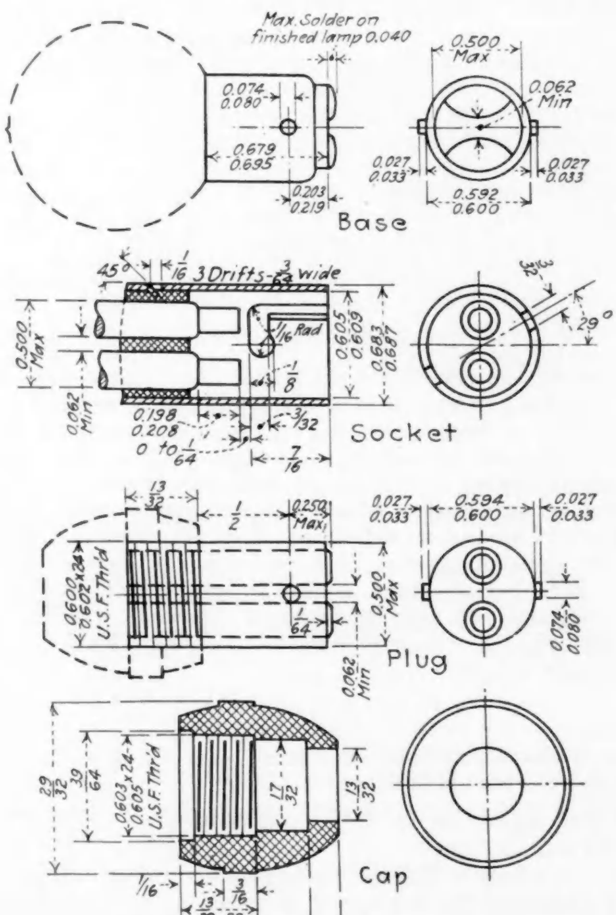
Plugs.—Change the thread limits for the plugs from 0.603 to 0.605 in. to 0.600 to 0.602 in. to permit better assembly, as a maximum-size plug is too large to properly fit a minimum-size cap.

Change the projections of the brass inserts from 1/32 to 1/64 in. to permit the plug to turn into the connector without interfering with the plungers of the insulated-return or two-pole type.

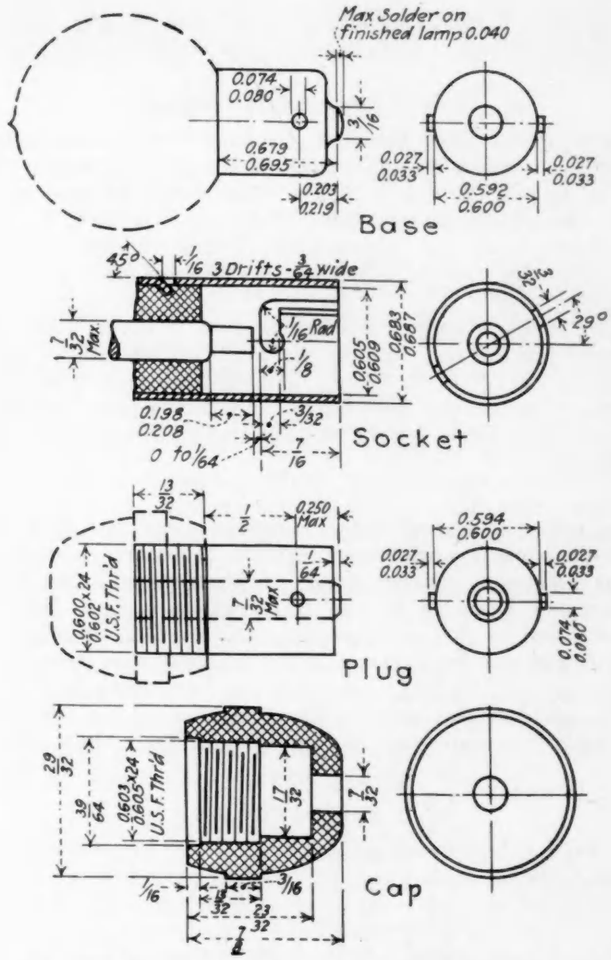
Sockets.—Change the length of the plungers from 0.281 in. minimum to 0.198 to 0.208 in. in order to conform to present practice and to eliminate unnecessary travel of the plungers.

Change the location of the bottom of the bayonet slot with relation to the ends of the plunger to limits 0 to 1/64 in. to conform to the shorter plungers.

Change the angle of the center-line of the slots to the center-line of the pins when locked from 39 to 29 deg.



INSULATED-RETURN TYPE BASES, SOCKETS AND CONNECTORS



GROUND-RETURN TYPE BASES, SOCKETS AND CONNECTORS

to conform to present practice and prevent die breakage.

Include the following recommended method of assembling the core in the tube: Three drifts shall be used and shall be approximately 3/64 in. wide, 1/16 in. long and drifted in at an angle of 45 deg. All drifts shall be made laterally and at one operation to prevent distorting the tube.

The present S.A.E. Standards for the ground-return and insulated-return type bases, sockets and connectors, revised as proposed above, are given in the accompanying illustrations.

PARTS AND FITTINGS DIVISION REPORT

(15) *Exhaust Pipes*

The Parts and Fittings Division's recommendation for S.A.E. Recommended Practice was approved in accordance with the following:

> The outside diameters of exhaust pipes extending from the engine to the muffler shall conform to the following steel-tubing sizes: 1, 1¼, 1½, 1¾, 2, 2¼, 2½, 2¾, 3, 3¼, 3½ and 4 in.

This recommendation was formulated in order to reduce the large number of exhaust-pipe diameters used in present practice and make it unnecessary for the muffler and exhaust-heater manufacturers to provide for an unnecessarily large number of sizes. Such a standard will be appreciated by the tubing mills and will facilitate deliveries and stocking. The recommendation is intended to apply to future production and is based on a survey

of present practice which indicated that the 1⅝, 1¾, 2, 2¼, 2½, 2⅝ and 3-in. steel-tubing sizes are most generally used.

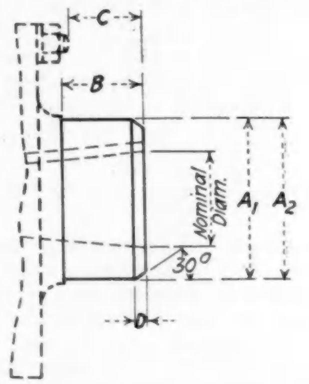
(16) *Square Shaft Fittings*

The Parts and Fittings Division's recommendation that the present S.A.E. Standard for Square Shaft Fittings, page C12, Vol. I, S.A.E. HANDBOOK, be revised so that the shaft-end and nut dimensions conform to the shaft-end and nut dimensions of the present S.A.E. Standard for Taper Fittings with Plain and Slotted Nuts, was approved.

The latter standard was revised in August, 1920, in order that the nut dimensions and threads might conform to the S. A. E. Standard regular thread pitch and U. S. Standard nut dimensions and thus permit the use of standard wrenches as well as facilitate the purchase of stock.

(17) *Universal-Joint Hubs*

In order to obtain better interchangeability between universal-joints and transmissions, the Parts and Fittings Division recommended adoption as S.A.E. Recommended Practice of the universal-joint hub dimensions given in the accompanying table. The dimensions specified are of importance to the transmission manufacturers by eliminating many present variations in these parts, and to the transmission purchasers by making available transmissions having standardized shaft-ends.



Nominal Diam.	Hub Diameter: Lathe Finish (A_1) Max.	Lathe Finish (A_1) Min.	Ground Finish[3] (A_2) Max.	Ground Finish[3] (A_2) Min.	Minimum Finished Length (B)	C[4]	D
¾	1.280	1.270	1.253	1.250	⅝	9/16	1/32
⅞	1.530	1.520	1.503	1.500	¾	⅝	3/64
1	1.780	1.770	1.753	1.750	¾	¾	1/16
1⅛	2.030	2.020	2.003	2.000	⅞	¾	1/16
1¼	2.155	2.145	2.128	2.125	⅞	¾	1/16
1⅜	2.280	2.270	2.253	2.250	1	¾	3/32
1½	2.530	2.520	2.503	2.500	1	¾	3/32
1¾	3.030	3.020	3.003	3.000	1¼	15/16	⅛
2	3.280	3.270	3.253	3.250	1½	1 3/16	⅛

All dimensions in inches.

[3]When specified the maximum eccentricity of the ground surface with respect to the hole shall be 0.002 in. (indicator reading 0.004 in.).

[4]The transmission design should provide clearance for the least distance from the end of the hub to the end of the flange bolt.

All fittings shall be S. A. E. Standard taper or spline fittings. The nominal diameter applies in either case.

The original recommendation of the Division, which was circularized among the universal-joint, transmission, passenger-car and motor-truck manufacturers, specified dimensions for the outside diameters of the universal-joint companion flange in order to permit the transmission designer to lay out the gearshift rods so that they would clear the companion flange. Criticisms received, however, indicate that the large number of different types and sizes of companion flanges would not permit specifying outside diameters without affecting individual design.

It is planned to have the Parts and Fittings Division formulate a standard for a complete series of flexible-disc couplings, which will be of value in this connection as well as for other purposes.

Screw-Thread Division Report

(18) *Drain-Cocks*

As radiator manufacturers and users have experienced trouble in draining radiators, due to sediment stopping up the small passage in many drain-cocks, and as the diameter of the passage varies considerably for the same size cocks, the subject was assigned to the Screw-Thread Division for consideration at the request of the Radiator Division. It was believed the diameter of the drain-cock passage for the different connection thread sizes should be definitely standardized.

The Screw-Thread Division secured data from the drain-cock manufacturers and the following recommendation for S.A.E. Recommended Practice was approved. The recommendation applies to drain-cocks intended for general use where these sizes are applicable, and is not limited to radiator applications.

Thread Size[5]	Passage Diameter, Minimum[6]
⅛	5/32
¼	1/4
⅜	11/32
½	7/16

[5]American Standard Taper Pipe Thread.

[6]The sectional area of the passages throughout their length shall be not less than that obtained with the minimum diameters given in the accompanying table.

Stationary Engine Division Report

(19) *Stationary Engine Belt Speeds*

The Stationary Engine Division has considered the standardization of belt speeds for stationary engines and power-driven implements, one of the most important subjects before it. A survey of present practice was made which indicated variations of belt speeds used by different manufacturers of engines and implements, these variations having an extensive effect on pulley manufacturers also. A number of horsepower-belt-speed curves were developed in order to have a definite belt speed for each engine size, and the present proposal was finally arrived at. The curve of these values is developed by the hyperbolic formula

$$(x - 400)^2 = 160,000\ (y - 1\tfrac{1}{2})$$

where

x = Nominal engine horsepower
y = Belt speed in feet per minute

and lies very close to the averages of present practice throughout its range.

The Stationary Engine Division's recommendation of the adoption of the belt speeds given in the accompanying table for stationary internal-combustion engines and for power-driven agricultural implements was approved.

Belt speeds are not specified above 16 hp. as there is a

tendency toward using tractor power for power-driven machinery requiring more than 12 hp. to which the present S.A.E. Standard tractor belt speeds of 1500, 2600, 3000 and 3500 ft. per min. apply.

STATIONARY ENGINE BELT SPEEDS

Nominal Engine Rating, Hp.	Belt Speed, Ft. per Min.	Nominal Engine Rating, Hp.	Belt Speed, Ft. per Min.
1½	550	7	1340
1¾	600	8	1420
2	680	9	1500
2½	800	10	1575
3	900	12	1700
4	1030	14	1810
5	1150	15	1860
6	1250	16	1910

THE DISCUSSION

MR. MENGES:—In connection with the report on belt speeds 23 different manufacturers were consulted. These speeds are the average of those used at present. An attempt was made to keep speeds within certain limits, but as it was found impossible to do so, the best thing under the circumstances was to take the average belt speed for each size stationary engine.

G. W. GILMER:—Transmitting power by belt is one of the oldest subjects that this Division has taken up. It is, I believe, one about which there is little general information. There are two generally accepted ways of compensating for changes of power in a belt. One is by changing the diameter of the pulley, thereby changing the speed, and the other is by changing the width of the belt. The latter is not in accordance with the best and most economical method of transmitting power. There are certain speeds at which engines operate most economically and satisfactorily. With reference to general power transmission, belt speeds between 1800 and 2500 ft. per min. are most desirable. If belt widths can be kept the same, and pulley diameters changed so as to maintain these belt speeds, approximately much better results will be had.

Belts running at low or very high speeds are not economical. None of the belt speeds given in the report comes up to the most economical speed for power transmission. I am a manufacturer of power belting and have been studying the subject for a number of years, longer than I have been a member of this Society, and I feel that if the speeds recommended by the Division are established, it will be necessary to change them in the future.

MR. MENGES: — We did not take into consideration what speeds would be the best and most efficient for the belts. The speeds recommended are an average of those used at the present time by the different engine builders. So far as power is concerned, it must be taken into consideration that the smaller engines drive slow-moving machines, are cranked usually by hand and operate at from 60 to 80 r.p.m. Under such conditions it is not possible to put on the most efficient belting, as it would be so narrow and run at such high speeds as to not be practical for the farmer.

(20) *Stationary Engine Rating*

The Stationary Engine Division's recommendation that the accompanying horsepower formula be approved only as general information for publication in the S.A.E. HANDBOOK, as a means of securing uniform practice in commercially rating stationary engines, was acted upon favorably by the Standards Committee and the Society.

$$\text{Nominal Engine Horsepower} = \frac{0.7854\, D^2 L R N}{13{,}000}$$

where
D = the piston diameter in inches
L = the stroke in inches
R = the revolutions per minute of the crankshaft
N = the number of cylinders

The formula is based on a piston displacement of 13,000 cu. in. per min. per hp., which is considered a very fair average factor for stationary and tractor engines burning either kerosene or gasoline. Various mechanical arrangements and refinements will, of course, influence the actual results on any one engine.

When the Tractor Division was originally working on this subject in 1919, the then tentative proposal, was submitted to and approved by the Stationary Engine Division as a satisfactory commercial or comparative rating for stationary internal-combustion engines, but failed of approval by the Standards Committee in June 1920 for adoption as S.A.E. Recommended Practice as being of a commercial rather than an engineering nature. It was approved, however, as general information.

The results obtained using this formula are almost exactly 80 per cent of the brake horsepower under average good conditions and provide the desired 20 to 25 per cent of reserve power. It is not intended that this formula shall be used in engineering calculations.

(21) *Lubricator Cups*

Members of the Stationary Engine Division have felt the need of a standard for stationary internal-combustion engine lubricator and, after obtaining and comparing data from the several manufacturers, the Division has formulated a series of lubricator cups, including the capacity, outside glass diameter, glass height and connection thread, which through its adoption will bring practice into accord. The report as recommended by the Division for S.A.E. Recommended Practice is given in accompanying table as approved by the Standards Committee.

LUBRICATOR CUP DIMENSIONS

Size No.	Capacity, Oz.	Outside Diameter of Glass, In.	Height of Glass, In.	Thread[7]
1	1	1½	1⅜	¼-18
1½	1½	1¾	1⅝	¼-18
2	2½	2	1⅞	⅜-18
3	4	2¼	2⅛	⅜-18
4	5	2½	2⅜	⅜-18
5	10	3	3	½-14
6	18	3½	4	½-14
8	32	4¼	5	¾-14

[7]American Standard Taper Pipe Thread.

(22) *Stationary Engine Crankshafts*

It is felt by some of the stationary-engine builders that standard crankshaft and crankpin diameters will be of material benefit in reducing the number of sizes of shaft-mounted pulleys, bearings and other parts affected by these diameters, facilitate purchasing and machining of crankshaft stock, and enable engine manufacturers to make replacements more readily. A survey of present practice indicated that these dimensions are nearly the same for the different makes of engines in corresponding engine sizes and that the proposed standard can be readily adopted by the manufacturers in new designs.

In some instances alloy steels have been used which have permitted of reducing the shaft sizes to a certain extent, but in general present practice is to use a low-carbon steel as service conditions necessitate a rugged low-priced, reliable construction.

The Stationary Engine Division's recommendation given in the accompanying table was approved for adoption as S.A.E. Recommended Practice.

CRANKSHAFT AND CRANKPIN DIAMETERS FOR SINGLE-CYLINDER STATIONARY INTERNAL-COMBUSTION ENGINES

Cylinder Bore	Crankshaft and Crankpin Diameter	Cylinder Bore	Crankshaft and Crankpin Diameter
3	1⅛	5½	2⅛
3¼	1¼	6	2¼
3½	1¼	6½	2⅜
3¾	1⅜	7	2½
4	1½	7½	2¾
4¼	1⅝	8	3
4½	1¾	8½	3¼
4¾	1⅞	9	3½
5	2	9½	3¾
...	...	10	4

All dimensions in inches.

(23) *Water-Pot or Hopper Capacities*

As it is believed that the volumetric capacities in present stationary-engine water-pot practice is in general close enough to warrant the adoption of a definite recommended practice at this time, the Stationary Engine Division recommended that the accompanying table of water-pot or hopper capacities be adopted as S.A.E. Recommended Practice. Although there is some difference in these capacities for kerosene and gasoline engines, they are considered near enough alike to warrant but one series of standard capacities for all practical purposes. The accompanying table was accordingly approved by the Standards Committee.

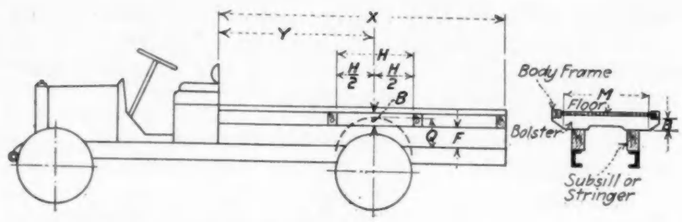
WATER-POT OR HOPPER CAPACITIES

Nominal Engine Rating, Hp.	Cooling Water Capacity, Lb.	Nominal Engine Rating, Hp.	Cooling Water Capacity, Lb.
1½	15.0	7	85.0
1¾	17.5	8	105.0
2	20.0	9	130.0
2½	25.0	10	160.0
3	30.0	12	225.0
4	40.0	14	310.0
5	55.0	15	360 0
6	70.0	16	420.0

TRUCK DIVISION REPORT

(24) *Motor-Truck Bodies*

The subject of motor-truck body-installation dimensions was first proposed to the Society about two years ago and assigned to the Truck Division. The original plan was to develop a standard which would include dimensions for the mounting of bodies on trucks and also frame widths, distance between spring-centers, wheel treads and turning radii. A Subdivision was appointed which obtained considerable data on the practice at that time. The work was delayed somewhat due to conditions which arose, and it was not until last fall that further data were secured and another careful analysis of existing conditions made. The original program has been reduced so that this proposal includes only dimensions



F (Subsill) = (*Q* + Chain Clearance) — *B*

Nominal Capacity, Tons	Body Bolster (B)	Length Back of Seat to End of Frame (X)	Length Back of Seat to Center-Line of Rear Axle (Y)	H[8]	Dimension of Bolsters (M)
¾ to 1	5	108	60	32	36
1¼ to 1½	5	120	72	36	38
2 to 2½	5	132 156	81 98	36	38
3 to 4	5	144 192	90 114	36	42
5 to 6	5	144 192	90 114	36	42

[8]Dimension H permits a variation of plus or minus 2½ in. of the rear axle from the normal position.

Q is measured from the top of the chassis to the top of the tires when the springs are deflected to the "metal to metal" position.

for the stringers, bolsters, distance from the back of the seat to the center of the rear axle and the distance from the back of the seat to the rear end of the frame. The dimensions were generally approved as suitable for the interchangeable mounting of truck bodies with the exception of special types such as large van bodies and special short dump bodies.

The proposal as submitted by the Truck Division for adoption as S.A.E. Recommended Practice and approved by the Standards Committee is shown in the accompanying drawing and table, and indicates the available space in which bodies may be mounted. The proposal also includes the nomenclature of the body mounting timbers, which at the present time are known by a variety of terms. It is believed that the proposal includes all the necessary dimensions for the installation of truck bodies by body builders on trucks conforming closely to the dimensions given in the table, and that it will answer the requirements for such standardization which had been urged through the technical press for some time.

(25) *Motor-Truck Hubs*

In April, 1919, the Truck Division believed that the standardization of wheel hubs for motor-trucks should be taken up for the benefit of truck operators by making possible interchangeability of wheels in emergencies, and permitting a reduction in the variety of axle bearings and hub dimensions, and for the benefit of the parts makers. A general questionnaire was sent out by the Society. The Wood Wheel Manufacturers Association and the Automotive Metal Wheel Association have cooperated very effectively with the Society throughout this program of standardization. A cooperative committee comprising representatives of the axle manufacturers prepared a tentative proposal which was submitted to axle bearing and vehicle manufacturers for comment. This proposal included five sizes of motor-truck front-axle spindles with their bearings and hubs laid out in considerable detail. The proposal, modified somewhat by reason of comments received, was reviewed at a meeting of axle manufac-

turers and changed slightly, after which it was considered by the Truck Division, and a report presented to the Standards Committee Meeting last January. After extensive and detailed discussion of the proposal, it was returned to the Truck Division for further consideration in accordance with the discussion at the Standards Committee Meeting. The Truck Division held a meeting immediately thereafter, at which a program was decided upon, and a resolution passed and published in THE JOURNAL to the effect that the Truck Division would reconsider the proposal, and desired further information in this connection from the industries.

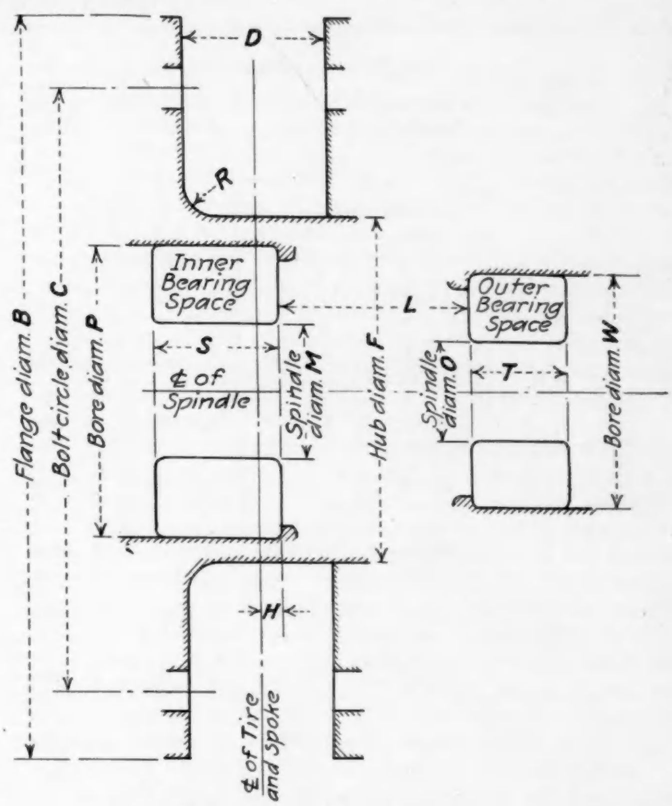
A meeting was called on March 3 in Detroit, which was attended by representatives of parts and vehicle manufacturers interested in this program. The proposal up to this time had included dimensions for inch-size roller-bearing installation only, but it was strongly believed that provision should be made in a standard for ball bearings also. A committee was thereupon appointed, the members of which were representative of the bearing manufacturers, with the understanding that if it was found feasible to incorporate dimensions for ball bearings in the proposed standard, definite recommendations would accordingly be submitted by the committee. This committee met in Cleveland on April 4, and submitted its recommendation that provision be made in the proposed standard for definite ball bearing sizes of suitable capacities for the proposed series of hubs, these ball bearings being interchangeable with the present corresponding S.A.E. Standard metric roller bearing sizes.

A meeting of the original committee appointed by the wheel manufacturers was held in New York City on April 19 at which the recommended proposal was reviewed in some detail before the whole matter was finally considered by the Truck Division. The Truck Division held its meeting in New York City on April 21, a number of representatives of the vehicle, axle, bearing, and wheel interests being present. The whole proposal as presented to the Truck Division included two series of spindle sizes with their related parts in considerable detail. After thorough discussion of this proposal, it was deemed best to limit the proposed standard to only such dimensions as are essential for the interchangeability of bearings and wheels on the proposed series of spindles.

The Truck Division's recommendation of the dimensions shown in the accompanying drawing and tables was then submitted to the Standards Committee and approved for adoption as S.A.E. Recommended Practice.

TABLE 6—DIMENSIONS FOR INCH SIZE TAPER ROLLER BEARING HUBS

		Hub and Spindle Number				
	Letter	R5	R6	R7	R8	R9
Diameter of flange	B	9	9¾	10½	10¾	11¾
Flange fillet radius	R	½	½	¾	¾	¾
Diameter of bolt circle	C	7¼	8	8¾	9¼	10¼
Number of flange bolts	..	12	12	12	12	12
Diameter of flange bolts	..	½	½	½	⅝	⅝
Spoke thickness between flanges	D	1¾	2	2¼	2½	2¾
Hub diameter for wheel bore	F	4.1250 4.1280	4.6250 4.6280	5.1250 5.1290	5.5630 5.5670	5.9380 5.9430
Inner edge of inner bearing to centerline of spoke	H	¼	0	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{11}{16}$
Inner bearing shoulder to outer bearing shoulder	L	2¼	2⅞	2⅝	3½	3 ½
Spindle diameter at inner bearing	M	1.7495 1.7485	1.9995 1.9985	2.1245 2.1235	2.4995 2.4985	2.5620 2.5610
Spindle diameter at outer bearing	O	1.1870 1.1860	1.4995 1.4985	1.7495 1.7485	1.9995 1.9985	1.9995 1.9985
Hub bore for inner bearing	P	3.4820 3.4810	3.9810 3.9790	4.3720 4.3700	4.7310 4.7290	5.1840 5.1820
Hub bore for outer bearing	W	2.8570 2.8560	3.1540 3.1530	3.4820 3.4810	3.9810 3.9790	3.9810 3.9790
Overall length of inner bearing	S	1.5000	1.5000	1.5000	1.7500	2.1250
Overall length of outer bearing	T	1.1875	1.1563	1.5000	1.5000	1.5000



MOTOR-TRUCK FRONT-WHEEL HUB DIMENSIONS

TABLE 7—DIMENSIONS FOR BALL BEARING OR METRIC ROLLER BEARING HUBS

		Hub and Spindle Number				
	Letter	B5	B6	B7	B8	B9
Diameter of flange	B	9	9¾	10½	11¼	12¼
Flange fillet radius	R	½	½	¾	¾	¾
Diameter of bolt circle	C	7¼	8	8¾	9¼	10¼
Number of flange bolts	..	12	12	12	12	12
Diameter of flange bolts	..	½	½	½	⅝	⅝
Spoke thickness between flanges	D	1¾	2	2¼	2½	2¾
Hub diameter for wheel bore	F	4.6250 4.6280	5.0000 5.0030	5.5630 5.5670	5.9380 5.9430	6.3750 6.3800
Inner edge of inner bearing to centerline of spoke	H	$\frac{5}{16}$	¼	⅞	$\frac{13}{16}$	⅞
Inner bearing shoulder to outer bearing shoulder	L	$2\frac{1}{16}$	2⅜	2¼	2⅞	2⅞
Length of ball bearing spacer	a	2⅝	2⅞	$2\frac{13}{16}$	$3\frac{9}{16}$	$3\frac{11}{16}$
Spindle diameter at inner bearing	M	1.7703 1.7689	1.9671 1.9657	2.1640 2.1626	2.3608 2.3594	2.5577 2.5563
Spindle diameter at outer bearing	O	1.3766 1.3752	1.5734 1.5720	1.7703 1.7689	1.9671 1.9657	2.1640 2.1626
Hub bore for inner bearing	P	3.9362 3.9340	4.3299 4.3277	4.7236 4.7214	5.1173 5.1141	5.5110 5.5078
Hub bore for outer bearing	W	3.1488 3.1466	3.5425 3.5403	3.9362 3.9340	4.3299 4.3277	4.7236 4.7214
Overall length of inner bearing	S	$1\frac{9}{16}$	1¾	$1\frac{15}{16}$	2⅛	$2\frac{5}{16}$
Overall length of outer roller bearing[9]	T	1⅜	$1\frac{7}{16}$	$1\frac{9}{16}$	1¾	$1\frac{15}{16}$

[9]When a wide type outer hub ball bearing is used, the spacer length should be L + $\frac{1}{16}$ in.

TABLE 8—INCH SIZE ROLLER BEARINGS

Spindle No.	Bock, Bower, Gilliam, Timken			Proposed Width	Manufacturers Number and Width of Bearings			
	Bearing	Bore	Outside Diameter		Bock	Bower	Gilliam	Timken
R 5	Inner	1.7500	3.4843	1.5000	(435-43) 1.5000	(435T) 1.5000	(435-4320) 1.5000	(435-4320) 1.5000
	Outer	1.1875	2.8593	1.1875	(3191-3110) 1.1875		(3191-312) 1.1875	(3191-3120) 1.1875
R 6	Inner	2.0000	3.9843	1.5000	(455-45) 1.5000	(455T) 1.5000	(455-4520) 1.5000	(4580-4520) 1.5000[11]
	Outer	1.5000	3.1562	1.1563	(3381-3310) 1.1875[10]		(3381-3320) 1.1563	(3381-3320) 1.1563
R 7	Inner	2.1250	4.3750	1.5000	(539-53) 1.5000	(539T) 1.5000	(539-532) 1.5000	(539E-532) 1.5000
	Outer	1.7500	3.4843	1.5000	(435-43) 1.5000	(435T) 1.5000	(435-4320) 1.5000	(435-4320) 1.5000
R 8	Inner	2.5000	4.7343	1.7500	(5564-5500) 1.7500	(5564T) 1.7500	(5564-553) 1.7500	(5584E-5520) 1.7500
	Outer	2.0000	3.9843	1.5000	(455-45) 1.5000	(455T) 1.5000	(455-4520) 1.5000	(4580-4520) 1.5000
R 9	Inner	2.5625	5.1875	2.1250	(63-6310) 2.1250			(6379-6321) 2.1250
	Outer	2.0000	3.9843	1.5000	(455-45) 1.5000	(455T) 1.5000	(455-4520) 1.5000	(4580-4520) 1.5000

All dimensions are in inches.
[10]Bock will decrease this width to 1.1563 in.
[11]New size bearing.

It is now proposed to continue this work to include a series of similar hub dimensions for passenger-car front-axle hubs and also, if feasible, for a complete line of rear-axle hubs for passenger cars and motor trucks.

The following reference tables of bearing sizes are printed with the approved report of the Truck Division as information in connection therewith but were not included in the report as submitted by the Division. The dimensions in Tables 6 and 7 of the Division's report refer to the hub and spindle diameters, while Tables 8 and 9 give the sizes of bearings which are applicable for each series of hubs and spindles.

TABLE 9—BALL BEARINGS AND METRIC ROLLER BEARINGS

Regular ball bearing installations are the Wide Type (double-row) inner and Medium Series (single-row) outer bearings.

Wide Type (double-row) outer ball bearing installations are considered special.

Spindle	Ball Bearings (S. A. E. Numbers)		Metric Roller Bearings[13] (S. A. E. Numbers)	
	Inner	Outer[12]	Inner	Outer
B 5	309	307	Rm 309	Rm 307
B 6	310	308	Rm 310	Rm 308
B 7	311	309	Rm 311	Rm 309
B 8	312	310	Rm 312	Rm 310
B 9	313	311	Rm 313	Rm 311

[12]For Medium Series (single-row), see S. A. E. Handbook, Vol. I., page C 28; for Wide Type (double-row) see page C 31.
[13]S. A. E. Handbook, Vol. I., page C 44.

UNACCEPTED RECOMMENDATIONS

Isolated Electric Lighting Plant Division Report

Rating of Storage Batteries

The report of the Isolated Electric Lighting Plant Division on the subject of proposed revision of rating of storage batteries, which includes only that portion of the report which was approved, is printed on page 63 of this issue of The Journal. The following part, printed as Section 2 in the report of the Division is given below together with the discussion on the whole report. The changes desired by the lighting-plant manufacturers at the subsequent meeting of the Gas Engine and Farm Power Association are printed in italics and enclosed in brackets.

SECTION 2

Storage batteries for farm light and power purposes shall be rated in terms of the number of hours discharge capacity at a constant rate corresponding to 300 [*200*] watts, or fifteen [*ten*] 20-watt lamps.

In determining isolated electric light and powerplant battery ratings, manufacturers shall conform with the following conditions:

(1) The normal range of specific gravity which is recommended by the battery manufacturers for the batteries in service shall be used during tests.
(2) Battery ratings shall be established at an initial temperature of cells not to exceed 80 deg. fahr.
(3) The watts at which the rating of lead batteries is determined shall be based on a normal voltage of 2 volts per cell. The final voltage on continuous dis-

charge shall not be less than 1.75 volts per cell.

(4) The batteries to be tested shall not be charged more than 120 per cent (in ampere-hours) of the last previous discharge

(5) The resultant test shall indicate the number of hours of service lead-acid batteries will give when discharged at a constant rate corresponding to 300 [*200*] watts.

[(*6*) *At 200 watts, 32 volts, the constant discharge rate shall be 6.25 amp.*]

THE DISCUSSION

MR. KEILHOLTZ:—Two years ago, at the summer session at Ottawa Beach, the Society adopted the intermittent battery rating for farm lighting plants and since then there has been considerable discussion of the subject. The lighting plant manufacturers, who form a section of the Gas Engine and Farm Power Association, voted at their convention in Chicago last September to request the Society to change this intermittent rating to the 8-hr. rating. Since then, the Isolated Electric Lighting Plant Division of the Standards Committee has met and considered a new method of rating expressed in watt-hours or kilowatt-hours. At a later meeting it was decided to recommend a continuous-discharge rating corresponding to 300 watts. Then this proposal was referred to a meeting of the lighting plant manufacturers section of the Gas Engine and Farm Power Association, which requested that the 300-watt continuous-discharge rating be changed to 200 watts.

Now it is proposed to abandon the present standard intermittent battery-rating as stated in the first section of the report. In the second section the proposed new rating for batteries is given in terms of the number of hours discharge capacity at a constant rate corresponding to 300 watts, or equivalent to fifteen 20-watt lamps. This was to be determined in accordance with certain specifications which are listed in the report.

Five and a half years ago, when the Delco Light Co. was organized, and it was the first to make large quantities of unit lighting plants, there was no standardized method of rating batteries; so it adopted what has been since known as the intermittent battery rating. This was later adopted by the lighting plant manufacturers. Later on, however, they wished to change, and since then they have first desired one kind of a rating and then another.

The proposed rating is not as satisfactory for small batteries as for large batteries, while the present intermittent rating is fair to both large and small batteries. Another point is that the proposed rating is not altogether fair to thick-plate batteries. A thick-plate battery cannot discharge as quickly as a thin plate battery, and it needs a discharge rating over a long period of time, such as the 72-hr. rating that is now the S.A.E. standard. This proposed rating is not satisfactory to the Delco Light Co., and we cannot see our way clear to adopt it. In our opinion it is a rating that nobody will use if it should be adopted, and I move that the intermittent battery rating be continued until a more satisfactory method can be found.

CHAIRMAN BACHMAN:—You have heard the report as presented by Mr. Keilholtz. He has made a motion which is contrary to the recommendations of the Division. I do not like to refuse anybody the opportunity to make a motion, but it places the chair in a rather embarrassing position to have the motion made in that way at this time.

MR. KEILHOLTZ:—Would you prefer that I withdraw the motion for the present?

CHAIRMAN BACHMAN:—I would, as I think it would leave the matter in a somewhat clearer condition.

MR. KEILHOLTZ:—I withdraw the motion.

CHAIRMAN BACHMAN: — You have heard Mr. Keilholtz's presentation of the report and also his personal views on the subject. This whole question of the rating of batteries is a rather important one. It will be recalled that several years ago, when arrangements were made whereby the Society became the authorized body for conducting the standardizing activities and engineering functions of a certain group of commercial organizations, it assumed a considerable amount of responsibility, and is obligated to discharge that responsibility as well as possible.

The Society contributed to those gentlemen who came from the organizations referred to a large amount of experience in handling standards work and an organization which was competent to handle it; but so far as personal knowledge of the individual problems of the isolated electric lighting plant manufacturers is concerned, a great many of us are connected with an almost entirely different line of activity, which probably makes it somewhat difficult for us to translate our own experience into terms of their experience. Therefore, those of us who are building automobiles, trucks, airplanes or motorcycles may not have a fully sympathetic realization of the problems which these gentlemen have confronting them.

The present standard rating which is printed in the S.A.E. HANDBOOK was adopted after serious consideration, but has met with much opposition in many quarters. Any remarks that I may make are open to rebuttal, but I want to clarify the matter in your minds. It may be that much of the opposition is more of a commercial than an engineering nature, and perhaps we are facing a situation with regard to this subject similar to one we faced on another subject a few years ago at Ottawa Beach.

In the endeavor to satisfy the requirements of those opposed to the present standard, further rather extensive work has been done. There has been voluminous correspondence on the subject, a number of meetings have been held, and another proposal has been made. It is Mr. Keilholtz's opinion, and he is not entirely alone in his position, that the present rating does not meet all requirements. It is his opinion that the present standard is better than the one now proposed. Unfortunately, many of us who are not thoroughly familiar with the situation are sitting as judges.

With this brief outline of the situation, I will depart somewhat from our usual method of procedure and invite discussion before we have a motion.

C. T. KLUG:—From a battery manufacturer's standpoint, the 8-hr. rating is desirable; it is a rating that has been used for many years. As I understand it, the farm lighting plant manufacturers wish a rating expressed in terms of watts or watt-hours. In a comparative test of batteries it makes little difference what the discharge rating is inasmuch as every one should use the same rating. I suggest a 320-watt discharge rate as a compromise, which is equivalent to sixteen 20-watt lamps, and at the same time equivalent to a 10-amp. discharge rate. In discharging a storage battery the amperage could be held constant not by a certain number of 20-watt lamps but through some fixed resistance, and I think if rating in watts is wanted, 320 watts would be better than 300.

L. W. HEATH:—I was present at all of the Division meetings held to consider this farm-light battery-rating

matter. Mr. Keilholtz was not present at the last two meetings and we held the last meeting of the Division over an additional day so that he could attend. For the benefit of those here who are members of the Standards Committee and have to vote on this question, I want to say a few words on the situation, and I may express them somewhat differently than Mr. Keilholtz did.

In the first place, the present standard intermittent rating has been found unsatisfactory by nearly all makers of farm lighting powerplants, so much so that at a meeting of the farm-light-plant manufacturers held last fall in Chicago it was voted to ask the Society to cancel the present rating. The matter was then taken up by the Isolated Electric Lighting Plant Division, which has held two meetings since then. A number of methods of rating were suggested, and finally at the last meeting a vote was taken that the Division would agree on some one rating. Before adjourning, those present went on record as being in favor of the rating now proposed. This proposal was then referred to a meeting of the farm-light-plant manufacturers held shortly thereafter and it was agreed that the proposed rating was satisfactory, except that the rate of discharge should be changed from 300 to 200 watts. There was not time to have another meeting of the Division before this meeting of the Standards Committee. Others are here who were at the meetings referred to, and I think they will bear me out when I say that a great majority of the farm-light-plant manufacturers are opposed to the present intermittent rating and in favor of the rating recommended in the report, with 200 watts specified as mentioned.

Mr. Chryst:—I may have my own personal opinions on this question, but I want to speak now from the standpoint of the Standards Committee. I believe that the Society should not have any standards which are continually being changed and disagreed with. Inasmuch as this report is submitted in two sections, first, that the present rating should be cancelled and, second, that a new rating be adopted, I move that the Standards Committee approve the cancellation of the present rating and refer the whole subject back to the Division for further consideration and general agreement, inasmuch as the present rating is unsatisfactory and the Division itself does not seem to agree on the proposed new rating.

Mr. Keilholtz:—I believe that we should not continue a rating that is unsatisfactory to at least the majority, but I am sure that the proposed rating would not be satisfactory. I think that this subject should be referred back to the Division for further work until it can decide on some battery rating that the lighting-plant manufacturers will approve.

Chairman Bachman:—We may be getting into difficulty if we vote on the two sections of this subject at once. If we vote on one section at a time we will have a much clearer conception of what we are doing. Can we have the motion withdrawn and a motion made on each of the two sections?

Mr. Chryst:—I move that the first recommendation, Section 1, to cancel the present standard, be adopted.

Chairman Bachman:—It is clearly understood that Mr. Chryst's motion is to the effect that the report of the Division relative to the cancellation of the present standard be adopted.

[Mr. Chryst's motion was duly seconded and carried.]

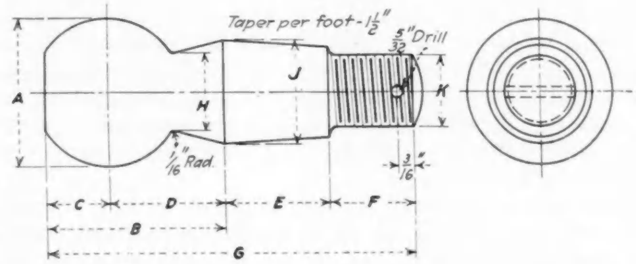
Chairman Bachman:—With regard to the second section of the Division's report, dealing with the proposed new rating at a 300-watt load changed to 200 watts, what is your pleasure?

Mr. Keilholtz:—I move that the Standards Committee do not approve the adoption of this Section 2, but refer it back to the Division.

[Mr. Keilholtz's motion was duly seconded and carried, there being one negative vote.]

Ball Studs

Early in 1920 the importance of standardizing ball studs was brought to the attention of the Society by parts manufacturers, who stated that ball studs are often more expensive than they would be if standardized, and that such a standard would be of value to manufacturers and users. The subject was assigned to the Miscellaneous Division and the data obtained by circularizing carefully reviewed. A recommendation was then formulated and circularized among motor-truck and axle manufacturers for comments which were carefully considered at a meeting of the Parts and Fittings Division to which the subject had been transferred. Several of the suggestions which had been received were incorporated in the recommendation of the Division as submitted for adoption as S.A.E. Recommended Practice and as given in the accompanying table.



No.	A	B	C	D	E	F	G	H	J	K
1	1	1⅜	7/16	15/16	13/16	13/16	3	⅝	0.766	⅝-18
2	1⅛	1½	½	1	13/16	13/16	3⅛	⅝	0.766	⅝-18
3	1¼	1 11/16	9/16	1⅛	1	13/16	3½	¾	0.875	⅝-18
4	1½	1⅞	⅝	1¼	1⅛	⅞	3⅞	13/16	1.000	¾-16
5	1¾	2⅛	¾	1⅜	1¼	1	4⅜	15/16	1.250	⅞-14

THE DISCUSSION

E. R. Douglas:—The company I am connected with has not been able to do much in general manufacture of ball studs because there have been so many kinds and sizes. Every car builder has his own sizes. We are very glad indeed that standardization is being attempted and want to see the best results attained. There are a number of particulars in which the design of ball studs should be considered. It is a little part, but one of considerable importance. While in general this proposal is very good, there are some points and some dimensions which should be reconsidered. In one size, for example, the diameter at the neck does not allow for enough angular motion of the socket on the stud. The other sizes are very good in that respect.

The strength of the ball stud involves several things; it involves material, which of course the manufacturer must choose; it involves the length from the center of the ball to the large end of the taper where it fits into the forging, and the diameters of the stud at *H* and *J*. With relation to the strength, ball studs of different sizes should have linear dimensions theoretically in strict proportion. That cannot be adhered to closely in prac-

tice because we do not want to machine all of the dimensions to thousandths of an inch but we can keep to practical proportions by adhering to ordinary dimensions. That proportionality is departed from, in a number of dimensions applying to the studs proposed, in one case rather widely. In that instance the departure gives a considerably greater strength in proportion to the load, and causes difficulty due to angular motion and some possible interference.

Room should be allowed for the chasers which cut the thread to run out but do not believe that a neck should be placed in the ball stud to allow the thread to run out. Dimensions which are necessary for those chasers to run out and for the use of standard nuts on the studs should be specified. The dimension F certainly does not allow sufficient room for a castellated nut.

There are one or two other points in the proposal which should have more consideration, and we believe the report should be returned to the Division for further study.

MR. HORINE:—With regard to the dimension A which is given as 1, $1\frac{1}{8}$, $1\frac{1}{4}$, $1\frac{1}{2}$ and $1\frac{3}{4}$ in., the balls must fit into the sockets, which are machined out by reamers that happen to come in the sizes specified. The balls should be slightly undersized, as the reamer is the part which we have to buy, that is, the tool which establishes the socket into which this ball must fit. For example, I recommend that on the No. 4 size the ball be 1 31/64 instead of $1\frac{1}{2}$ in. in diameter, that is, 1/64 in. under the reamer size. I also think that a radius of 1/16 in. between the ball and the neck, which is specified for all sizes of stud, is too small. That is the weakest point on the whole stud; on the larger sizes the 1/16-in. radius amounts to a real weakness. I suggest at least a 3/16-in. radius.

I believe that the length E is not great enough. It has been the experience of a number of manufacturers besides ourselves that studs fail because there is not sufficient length of taper. The $1\frac{1}{2}$-in. taper has been tried and found less satisfactory than the $1\frac{1}{4}$-in.

I suggest that this whole subject of ball studs be referred back to the Division for further consideration.

MR. STRICKLAND:—The report was based on data received from axle manufacturers located all over the country, and provides a reduction in the number of sizes in use today. There are, however, a few points that might be cleared up, and I think the report should be returned to the Division.

ADDRESS OF CHAIRMAN BACHMAN

In closing the session of the Standards Committee, Chairman Bachman addressed the meeting substantially as follows and conducted a brief executive session to give the members of the committee an opportunity to make suggestions in connection with the conduct of the work and to propose any new subjects to which the committee should give consideration.

We have had a shorter period of time than usual between the Annual Meeting and this one, and we have a correspondingly longer period of time between this meeting and the next Annual Meeting.

The reports that have been submitted today are highly representative of the quality of the standards work, and each time that we come together and consider reports, we increase our responsibility and our ability to get results. There is no question that one of the greatest activities the Society has ever gone into is automotive engineering standardization. It has been one of the things that has brought the Society to the favorable attention of not only the executives of our industry but other engineering and commercial organizations. I had the pleasure last week of addressing a commercial organization in Philadelphia on the subject of standardization, and was surprised that such a topic should have had the reception it had, particularly in the way that I could present it, and among a group of salesmen.

Now, I do not mean that standards work is without criticism. We have had some of it here today in our own meeting, and all the criticism that we can get of a constructive nature that is listened to and taken in the spirit in which it is given, will make for better work.

We have about 7 months before us until our next meeting. Of that 7 months we have probably three in which under even normal circumstances we would not hold meetings very often and do much work. Under the present conditions, however, we probably will be able to do even less. The fact that we have had such a good meeting today is I think a promising sign that we realize the need of cooperation and of coordinated effort. While we may not be able to see each other or to meet in committee, there are several things that we should do.

The reports which have been passed here today will be considered by the Council and very briefly at the business session tonight. It is impossible, under present conditions, to give very lengthy attention to any of these matters at the business sessions as the time is not available, but the Society office will send letter ballots for final approval of these reports by Society members. We have a large membership, and in the past only about 12 per cent of the voting members have returned these letter ballots on the Standards Committee reports. There is not a man in this room or a man in the Society who is competent to pass on every one of the subjects that is brought up in the Standards Committee, but a large majority are directly concerned in what the Standards Committee does in at least one or two matters which you and your friends know something about. Every voting member of the Society should return his letter ballot.

You who are members of the Standards Committee should constitute yourselves missionaries to see that those who are not directly connected with Standards Committee work give some attention to it, because only insofar as it is representative of the combined activity and thought of the members of the Society is it worth anything. If we do not get the benefit of such thoughtful cooperation it simply means that subjects come up, are reported and passed, that are deficient in one way or another, even with the consideration that we do give them. These deficiencies will eventually be published in the S. A. E. HANDBOOK and either be followed with disastrous results in one way or another, or detract from the reputation which we have to sustain.

The members of the several Divisions should take a direct personal interest in the work of their Divisions. Do not think that the chairman is the only one having any responsibility. Each member has been appointed by the Council, has accepted the appointment and has a direct responsibility to the Society as a whole and to the industry in general. Become familiar with the topics that have been brought up for discussion by the Division and give some constructive thought to them. Not every member can attend every Division meeting, but every one can give consideration to the subjects that are before the Division and submit his opinions in some definite form.

Finally, I want to touch on the question of having these matters properly circulated among those who are interested. We have an organization at New York City whose function and responsibility is to see that this is done. You cannot do it; I cannot do it; we are not supposed to, and the Council doesn't ask us to. It

does ask this office organization to do it, but it is very discouraging to prepare and send out inquiries and get very few acknowledgments and very little needed information. That is all I wish to say, but I believe it will be profitable for all of us to bear in mind t he necessity for helpful cooperation in this work in the future. If there are subjects which you think the Standards Committee or any of its Divisions should consider, send advices as to them to the Society so they may be properly assigned to a Division and work started in accordance with regular methods of procedure.

Attendance at Meeting

The members of the Standards Committee and the Society and the guests in attendance were

Standards Committee Members

J. J. Aull
B. B. Bachman
David Beecroft
R. S. Begg
W. J. Belcher
C. C. Bowman
A. E. Brion
W. M. Britton
A. K. Brumbaugh
H. E. Brunner
T. V. Buckwalter
R. S. Burnett
R. J. Burrows
C. C. Carlton
Clarence Carson
D. F. Chambers
W. A. Chryst
C. F. Clarkson
W. F. Cole
J. R. Coleman
A. W. Copland
C. S. Dahlquist
G. W. Dunham
E. H. Ehrman
J. B. Fisher
L. C. Fuller
C. E. Godley
C. O. Guernsey
F. W. Gurney
W. S. Haggott
L. W. Heath
M. C. Horine
E. A. Johnston
L. S. Keilholtz
W. C. Keys
C. T. Klug
G. L. Lavery
B. M. Leece
A. D. T. Libby
T. C. Menges
C. A. Michel
G. L. Miller
C. T. Myers
J. H. Nelson
I. M. Noble
H. S. Pierce
Gustof Peterson
F. R. Pleasonton
H. J. Porter
M. P. Rumney
C. E. Sargent
A. J. Scaife
A. W. Scarratt
C. W. Spicer
W. R. Strickland
J. G. Swain
G. J. Thomas
L. M. Wainwright
J. A. White
S. O. White
F. G. Whittington
Samuel Wilbur
Ernest Wooler
G. A. Young

S. A. E. Members and Guests

V. G. Apple
J. H. Baninger
R. C. Barron
G. M. Bartlett
Joseph Bijur
C. R. Bissell
W. J. Bryan
H. C. Buffington
W. T. Burns
O. E. Byron
Herbert Chase
H. E. Clay
E. O. Christiansen
K. H. Condit
L. R. Davis
E. Dickey
N. S. Diamant
G. P. Dorris
E. R. Douglas
F. G. Druar
W. H. Fenley
H. G. Freeland
G. W. Gilmer, Jr.
H. A. Githens
G. O. Hanshew
J. N. Heald
P. M. Heldt
L. C. Hill
M. G. Hillman
M. L. Hillmer
H. H. Hime
Russell Hoopes
H. W. Jarrow
Clyde Jennings
C. J. Kalbfell
L. M. Kanters
T. S. Kemble
W. H. Knowles
N. Lazarnick
W. E. Lay
R. G. McCaugha
John McGeorge
J. F. Marshall
J. A. Moyer
R. E. Northway
Joseph Pflum
N. B. Pope
W. I. Ralph
A. R. Reid
D. Roesch
O. J. Rohde
J. E. Schipper
E. H. Schwartz
C. F. Scott
F. A. Shuler
R. H. Soulis
C. R. Stough
A. L. Swank
W. S. Turner
A. S. VanHalteren
R. F. Walters
J. A. White
C. E. Wilson
H. G. Wilson
C. W. Wolfe
H. T. Wreaks
O. W. Young

BRAKE-LINING TESTS

THE development of satisfactory commercial tests for brake-linings is one of the problems with which the automotive engineer has been contending without having found an entirely satisfactory solution. Tests have been devised that were founded more or less on the individual's ideas, but have been of no practical value or shown any uniformity of methods.

At the joint meeting of the Truck Division of the Standards Committee and the Truck Committee of the National Automobile Chamber of Commerce held on April 21, 1919, discussion of the advisability of establishing standard tests for tractive ability, torque, brake efficiency and similar factors led to a decision to develop suitable brake-lining wear and heat tests for adoption and use in the determination of brake-lining specifications and coefficients of friction. It was suggested at that time that the work should be conducted by the Society in a laboratory of its own but this was obviously not feasible. As the Motor Transport Corps contemplated establishing purchasing specifications for brake-linings, arrangements were made for the Motor Transport Corps to furnish the necessary apparatus and the Bureau of Standards to conduct the tests, the work to proceed under the general supervision of a subdivision of the Truck Division in cooperation with the Motor Transport Corps and the Bureau of Standards. A. K. Brumbaugh, Clarence Carson and H. C. Dickinson comprised the Subdivision and in order that the tests might be conducted as rapidly as possible, arrangements were also made with the Bureau of Standards whereby the Subdivision furnished one of the engineers to perform the tests at the Bureau of Standards.

With the reorganization of the Standards Committee for 1921 this work, which was formerly assigned to the Truck Division, was transferred to the new Parts and Fittings Division. The original Subdivision was enlarged by adding representatives of four additional brake-lining manufacturers. A meeting was held in New York City on April 19, 1921, at which the results of the work accomplished up to that time were discussed in detail and the decision reached that each of the brake-lining manufacturers represented on the Subdivision would construct testing equipment similar to that installed at the Bureau of Standards and conduct an independent series of tests. It was felt that this would be of material assistance and afford an opportunity to obtain much comparative information.

At a conference called at the Bureau of Standards on May 17 which was attended by representatives of practically all the brake-lining manufacturers, a report and analysis of the progress which had been made there was submitted and discussed. Many valuable data and much information had been obtained by the Bureau which indicated that the problem is much more complex than was anticipated. It was felt generally that the testing should be continued and another conference called after the Bureau and the several manufacturers shall have more nearly completed their investigations.

The lack of definite information and uniform practice with regard to the testing and operation of brake-linings clearly indicates that the establishment of standard tests which can be followed by manufacturers and users and form the basis for purchase specifications will be an important accomplishment in this branch of the industry. Such standards when carefully planned and executed should make possible greater uniformity of materials and establish a better understanding of the essentials involved.

PRODUCTION ENGINEERING

THE production engineer is the man responsible for planning the routing of work through the shop, the sequence of operations and the maintenance of a regular flow of material to each machine, insuring that when the parts arrive at the erecting floor, all the necessary material will be on hand in correct quantities. Under the old methods the responsibility for laying out the progress of work through the shop rested with the shop foreman. With very rare exceptions his method of scheduling the work was to wait until a man finished one job before deciding which of the jobs waiting to be done should be undertaken next. This involved consideration on the part of the foreman as to which of the available jobs suited the particular man's ability and which of the jobs should be given preference. Frequently the waiting workman would be kept idle until the foreman had decided these points. Even after some particular job had been decided upon, more time would be spent in a discussion as to how the job should be handled.

This is not a criticism of the foreman because of his lack of planning as his duties under the old method were numerous and left him no opportunity for planning in advance. He was required to pass judgment on different questions as they arose, and upon his ability to give a snap judgment depended the success or failure of the whole matter. In the majority of cases, the output under the circumstances was very creditable. Under present conditions, however, it is impossible for even the smallest shop to run to the best advantage without more planning of the work than has been usual in the past. Therefore, in every shop it is necessary for someone to perform the duties of a production engineer. If the plant is small, this man will be able to perform additional duties, but it is essential that the problem of routing jobs through the shop be given consideration.

Production Engineering for the Small Shop

The scheduling of work through a small shop presents in some ways more difficult problems than it does in the large shop, owing to the relatively greater variety of work handled and the fact that there is a smaller number of men and of machines to choose from. For these reasons, the ability of the available mechanics becomes the vital point for consideration in distributing the jobs. In such shops it is necessary that whoever handles the scheduling must have an intimate knowledge of the ability of the men who are to do the work. The conclusion is, then, that in the medium-sized or small factory it is seldom that any one except the foreman would have the knowledge necessary to schedule the work successfully, but when thus burdened with production work he should be relieved of enough of his other less important duties to enable him to devote the necessary time to planning the work.

In actual experience the output of a moderate-sized machine shop has been increased more than 50 per cent by merely having the various jobs carefully planned ahead, choosing that work for each man at which he was most proficient, and having the next job with all essential information waiting at his machine before he had finished the previous job. In larger plants where a production engineer is employed, the work is scheduled to go to the machine that is most efficient for the particular job, and the foreman is responsible for obtaining the most suitable operator for the machine.

Method of Routing and Scheduling the Work

The end toward which the production engineer should work is to have every operation on every piece of work that is passing through the shop scheduled definitely and clearly both in regard to the machine which is to be employed and the time that it is to be performed; at least, the order in which the jobs are to be executed should be specified. The most desirable sequence of operations will usually be decided upon, and then, after estimating the time requirement for each operation, it is possible to distribute the work to the best advantage among the available machines so as to avoid overtaxing any of them. The best plan is to have the work arranged so that all of the machines will be kept busy all of the time.

The division of the work among the various machines demands careful consideration, especially when dies or special tools are required for executing the work. The reason for this is that generally, after the tools are made, it is difficult and expensive to transfer the work to another machine if it is found that the one chosen has been assigned more work than it can do according to schedule. The jobs should be run through in as large lots as possible to avoid changing tools too often, although some leeway must be had here, because if the lots are too large some unforeseen delay may prevent the delivery to the erecting gang of some essential part and thus stop work on the erecting floor. At the same time, this trouble is less likely to occur even with poor scheduling than it is with none at all.

It is surprising how many supposedly up-to-date plants are running the work through the shop with practically no planning, the infallible result being that some of the machines are loaded with a quantity of work beyond any possible output that can be expected of them, while other machines stand idle, thus seriously impeding the possible output of the plant. This condition usually results in a continuous rush to get out some part necessary to complete a job which may have been overlooked, and as a result the whole shop is thrown into a turmoil, trying to catch up with the erecting crew. To attempt to run any machine shop today without some kind of schedule is about as foolish as it would be to try to run a railroad without a time table. Furthermore, to wait until a man finishes one job before deciding on the next is on a par with a railroad on which the train crew has to stop at each station to find out at what place they stop next, if such a foolish thing can be imagined.

There are numerous reasons for employing engineering methods in directing the flow of production through the shop. While at first sight the labor and expense involved in maintaining a schedule may seem formidable, this is really not the case, since the economies that will be effected both in reduced costs and in an increased output will return a big dividend on the expenditure. It is not necessary that the work, to pay for scheduling, be repetition work, for it has been my experience that it pays as well to schedule the work through a repair shop as through a manufacturing plant. The schedule for a repair shop must naturally be more elastic than it is in the manufacturing plant to allow for the occasional breakdown job which must be handled at once, but even then the schedule has its advantages in that it gives a better idea of the situation and shows which work can be laid aside to take care of the emergency job. It may seem on first thought that the breakdown of a machine would throw the whole schedule out of order, but this is not true, as with the information that was used in making the schedule it is possible to rearrange it by transferring the work to some other machine, or find out just what it is possible to do at short notice more readily and surely than can be done without this information. The schedule also shows the amount of time that may be consumed in repairing the machine without holding up production.

Assigning Jobs for Maximum Production

It is common knowledge to those who have worked in the shop that a man will finish a job more quickly when he knows that the next one is waiting for him than he will when he does not know what he is to do next. This fact is especially noticeable when there happens to be a disagreeable job waiting to be performed that no one wants and every one tries to avoid until it has been assigned. Many machine operators make a habit of lagging in their work at the end of a job, thinking that it is entirely up to the foreman to keep them

busy, and as long as they can make an appearance of being at work they will not tell the foreman that they are ready for another job. Some will absolutely refuse to report when through with a job, but depend upon the foreman to know and to assign the next job when he gets around to deciding what it is to be.

This leaves all the responsibility of knowing just when each job is completed to the foreman. If a number of men finish their jobs at the same time, which often happens, a serious loss of time is incurred until the foreman can decide upon the next job for each, one after the other, and give each his instructions. Under these circumstances an error in judgment on the part of the foreman in distributing the work or in issuing instructions is very likely to occur. On the other hand, if a schedule were arranged, the foreman could distribute the work among the men without waiting for the completion of the various jobs.

The importance of and the knowledge required for production engineering cannot be over-emphasized. It demands a thorough knowledge of all machines, machining operations and time required to perform the work, as well as good judgment regarding general shop practice. The successful production engineer will not stop at the scheduling of the work, but will be able to assist the designing engineer in developing designs which can be most economically manufactured with the available equipment, thus avoiding the buying of new machines that cannot be kept in operation.—J. S. Watts in *Machinery*.

SPORTS CONTESTS AT THE SUMMER MEETING

THE customary track and field events that add to the interest of each Summer Meeting were held, under the direction of the Sports Committee, on the athletic field adjoining the hotel. All of the events were hotly contested and the winners deserved the useful and valuable prizes that rewarded their efforts.

The following summary, which was crowded out of the June issue of THE JOURNAL, lists the events and also includes the names of the prize winners:

50-yd. Dash, men under 30—first, K. M. Lane; second, E. O. Jones; third, L. H. Gaylord
50-yd. Dash, men 30 to 40—first, L. E. Joseph; second, Neil McMillan, Jr.; third, V. E. Clark
50-yd. Dash, men over 40—first, Max Tost; second, E. H. Schwartz; third, George J. Thomas
Fat Men's Race, 200 lb.—first, L. H. Gaylord; second, G. A. Sanford
Ladies Potato Race—first, Mrs. E. Dickey; second, Mrs. C. M. Day
Ladies Egg Race—first, Mrs. C. M. Day; second, Mrs. G. A. Kraus
Ladies 50-yd. Dash, under 25—first, Miss Ruth Porter; second, Miss Loretta Van Aman
Ladies 50-yd. Dash, over 25—first, Mrs. E. Dickey; second, Mrs. G. A. Kraus; third, Mrs. C. M. Day
Ladies Baseball Throwing—first, Miss Norma Porter; second, Miss Ruth Porter
Ladies Walking Race—first, Mrs. H. L. Horning; second, Mrs. G. A. Kraus
Men's Walking Race, under 35—first, Gordon Brown; second, S. E. Bates; third, H. M. Benstead
Men's Walking Race, over 35—first, V. E. Clark; second, David Beecroft; third, B. J. Lemon
Three-Legged Race—first, Neil McMillan, Jr. and K. M. Lane; second, E. O. Jones and L. P. Jones; third, L. H. Gaylord and W. F. Rockwell
Intersectional Relay Race—first, Detroit, B. W. Brodt, V. E. Clark, E. O. Jones, L. P. Jones.
Intersectional Relay Potato Race—first, Indiana, Mark Smith, captain
High Jump, men under 30—first, B. W. Brodt; second, K. M. Lane; third, M. P. Whitney
High Jump, men over 30—first, W. F. Rockwell; second, S. E. Bates; third, Charles Wolfe
Shot Put—first, H. E. Kirby; second, L. H. Gaylord; third, Gordon Brown
Standing Broad Jump, men under 30—first, M. P. Whitney; second, Gordon Brown; third, J. E. Padgett
Standing Broad Jump, men over 30—first, L. E. Joseph; second, F. S. Whittington; third, W. F. Rockwell
Running Broad Jump, men under 30—first, B. W. Brodt; second, M. P. Whitney; third, Gordon Brown.
Running Broad Jump, men over 30—first, F. S. Whittington; second, L. E. Joseph; third, C. W. Wolfe
Hop, Skip and Jump—first, B. W. Brodt; second, L. E. Joseph; third, J. G. Vincent; fourth, David Beecroft

In addition to the members of the Sports Committee, under the able leadership of its chairman, Howard A. Coffin, the following assisted in making the field day a success: B. G. Koether, head judge of track events; Mark Smith, head judge of field events; J. J. O'Neill, C. R. Bissel, clerks of the course, and C. B. Veal, starter.

IRON AND STEEL DIVISION MEETING

A WELL attended meeting of the Iron and Steel Division was held June 20 and 21 at the Society office. In the course of the meeting a conference was had with representatives of the Metallurgical Committee of the American Gear Manufacturers Association and action taken to add to the S. A. E. Standard Steels two additional types of steel intended primarily for gears.

W. C. Peterson, chairman of the Subdivision on Sheet Steel, submitted a report of progress which indicated a desire for cooperation on the part of the sheet steel manufacturers.

C. N. Dawe, chairman of the Subdivision on Specifications for Cast-Iron Valve-Heads, submitted a report, accepted as final, which indicated that it is practically impossible to develop a satisfactory specification of this kind at present.

R. M. Bird, chairman of the Subdivision on the Effect of Size on the Physical Properties of Steel, submitted a progress report including blueprints showing the results obtained with a limited variety of compositions.

The Division spent an entire day on the revision of the Notes and Instructions for the steel specifications that has been under way for the past year. Much progress was made. It is planned to hold another meeting of the Division at Niagara Falls during the week beginning July 24 to complete this work at that time if possible.

THE DAYTON AERONAUTIC MEETING

THE joint aeronautic meeting of the Society and the American Society of Mechanical Engineers, which was held at Dayton, Ohio, on May 21, proved most interesting. There were 200 members of the two societies in attendance and all joined in praising the Air Service staff at McCook Field for its success in arranging a program so diversified and complete. Fortunately weather conditions were ideal for flying and this enabled the pilots to demonstrate their individual skill as well as the flying qualities of the many airplane types that were ground-inspected by the visitors.

Upon arrival at the field the members were received by Major Thurman H. Bane, commanding officer and genial host, who conducted a tour of inspection through the laboratories and shops. The materials-testing, powerplant, wind-tunnel, propeller and armament laboratories were visited. Some 25 types of airplane engine were exhibited including several foreign models. The special equipment used in testing carbureters was demonstrated and the propeller dynamometer was shown in operation. Great interest was shown in the airplane-armament exhibit; several types of machine gun and cannon were fired from stands and also through a moving propeller by synchronizer control. Different forms of ammunition were used and the purpose of each demonstrated by firing into airplane tanks filled with gasoline.

After completing the tour of the shops, the members inspected the fleet of airplanes assembled on the flying field for the purpose. Opportunity is seldom offered for the study of so many different airplane types in one group, ranging from the large Caproni and Martin bombing planes to the little single-seater Messenger airplane. A representative lot of the war-time airplanes of German, French and British origin was displayed, as well as some of the more recent designs. The McCook Field pilots completed the field program with a thrilling display of stunting and aero-acrobatics. The members were very much amused by the uncanny antics of the small wireless controlled electric automobile which went through all manner of evolutions at the command of its operator who was stationed 75 yd. distant.

In the afternoon a series of 10-min. talks was given by members of the McCook Field engineering staff. E. H. Dix, Jr., described some of the development accomplished at the Field in aluminum-silicon alloys for air-cooled cylinder castings. The advantages of air-cooled aviation engines and the advances being made in their design were enumerated by S. D. Heron. A comparison between airplane and automobile radiators was made by Lieut. Bayard Johnson, who also described the core constructions commonly used. C. F. Taylor discussed the characteristics demanded of a carbureter for airplane engines and the means generally adopted to secure them. The interesting subject of airplane camouflage was covered briefly by G. P. Young who showed designs which render invisible above 10,000 ft. planes which otherwise could not fly under 17,000 ft. without being detected. The progress in wireless telephony and the perfection of the wireless direction-finder were outlined by O. E. Marvel who emphasized the important function of wireless apparatus in the commercial operation of air carriers. The most fascinating talk of the afternoon was that of Capt. G. W. Stevens on aerial photography. He illustrated various types of ingenious equipment devised during and since the war for taking successive interlapping exposures and constructing extremely accurate maps from them. It was surprising to learn how exactly a section of country can be reproduced and at a cost far less than that involved in the customary land-survey method. The impression was left by Captain Stevens that this development constitutes a most important commercial application of the airplane. H. O. Russell concluded the program with a description of the types of synchronizing mechanism used to control gunfire through propellers.

A dinner was given for the visitors by the Dayton Section of the Society at the Engineers Club in the evening. J. H. Hunt was toastmaster at this very enjoyable affair and introduced as speakers C. F. Kettering, J. A. Steinmetz and F. Handley-Page of England. They talked primarily of aviation of course but the members were equally interested in the keen wit of Messrs. Handley-Page and Kettering. The dinner was followed by a military ball given by the officers of McCook Field. This provided an opportunity for the Dayton ladies to entertain the visiting members, which they did admirably.

Thanks are due Major Bane and his fellow officers for their efforts in arranging the excellent program at McCook Field and to the Dayton Section of the Society for the very fine dinner. We know that the visit to Dayton was appreciated by all who made it and will not soon be forgotten.

RECENT COUNCIL MEETINGS

SESSIONS of the Council held at West Baden, Ind., on May 24, during the time of the Summer Meeting of the Society were attended by President Beecroft, Vice-Presidents Horning, Bachman, Crane, Johnston and Menges, Past-President Vincent and Councilors Germane, Pope and Scarratt.

One hundred and forty-three applications for individual membership, three for affiliate membership and three for student enrollment were approved. The following transfers in grade of membership were approved: Associate to Service Member, R. O. Eliason; Junior to Member, Charles Hollerith; Associate to Member, Robert W. Davis, William J. Foster, T. F. Cullen, P. C. Cloyd; Member to Service Member, W. B. Elston. One action taken which it seemed necessary to take, much to the regret of the Council, was to strike from the roster of the Society the names of 110 members who had not paid the dues for the fiscal year beginning Oct. 1, 1919.

It was reported that the National Automobile Chamber of Commerce had voted to continue its annual appropriation of $7,500 to the standardization and research work of the Society.

The following Standards Committee appointments were made, with assignment as indicated:

L. W. Close—Ball and Roller Bearings Division
J. B. Fisher—Chairman Engine Division
R. J. Broege—Vice-Chairman Engine Division
Louis Schwitzer—Engine Division
W. E. Perdew—Lubricants Division
Herschell G. Smith—Lubricants Division
J. R. Coleman—Parts and Fittings Division
W. E. Holland—Vice-Chairman Storage Battery Division
Albert R. Reid—Storage Battery Division
John Mainland—Vice-Chairman Tractor Division

The Standards Department was authorized to prepare a skeleton Division organization in connection with nomenclature work.

The following additional subjects were assigned for study by the Standards Committee: Truck Division, Dumping Hoist Platforms and Body Hold-Down Clamps; Screw-Thread Division, Taper Tools and Products; Tractor Division, Formula for Stability of Tractors.

E. A. Johnston, chairman, C. B. Rose, A. H. Gilbert, G. A. Young, R. O. Hendrickson and John Mainland, were named as a committee to cooperate with committees of the National Implement and Vehicle Association and the American Society of Agricultural Engineers in the formulation of agricultural equipment standards.

The change of the name of the Boston Section to New England Section was authorized.

A session of the Council was held at the offices of the Society on June 23.

ACTIVITIES OF THE SECTIONS

ONE of the most important events of the Summer Meeting was the Sections Luncheon held on Tuesday May 26 this being attended by members of the Council, officers of the Sections and members of the Sections Committee. Plans were discussed for improving the work of the Sections and very valuable suggestions were received.

President Beecroft called upon H. R. Corse, chairman of the Sections Committee, to outline the suggestions of that committee. The committee recommended that the program committee of each Section prepare within the next few weeks a tentative schedule of meetings, giving the proposed subject and author for each meeting during the coming year, and send this program to the office of the Society at New York for *suggested* additions or variations. The Committee believes that papers should be prepared sufficiently far in advance to enable them to be preprinted when desirable and sent to the Section members prior to the meetings at which they are to be presented. The discussion should under these circumstances be of much greater value.

R. E. Northway, past-chairman of the Boston Section, spoke of the desire of that Section to change its name to "New England" and the necessary change in the Section By-Laws was thereupon approved by the Council, a meeting of which was immediately called for that purpose.

H. W. Slauson, chairman, Metropolitan Section, agreed that the full season's program should be laid out in advance and promised that a list of subjects with hoped-for speakers could be obtained by early July. H. L. Horning told of the Mid-West Section's plans for the coming year which involve a series of lectures on thermodynamics, the principles of carburetion and combustion phenomena. The meetings will probably be held at Lewis or Armour Institute where physical laboratory apparatus is available. He spoke of the important work being done at Dayton on elimination of knock and said that vaporization of fuel is the most important problem today.

Lon R. Smith, chairman, Indiana Section, thought it desirable to coordinate the work of the different Sections to avoid duplication. Some members were of the opinion that repetition of a paper at more than one Section meeting is beneficial since added discussion results from such a plan. Dr. H. C. Dickinson, chairman, Washington Section, said that a number of meetings of a semi-popular nature had been held at Washington but that the program for this season would probably be of a more technical nature. T. F. Cullen, secretary, Pennsylvania Section, considered it possible to formulate plans for the season's work by July. He was interested in the proposed Aberdeen Proving Ground meeting. G. T. Briggs, vice-chairman, Mid-West Section, believed it advisable to hold inexpensive dinners in connection with the Section meetings and spoke of the success of the last Mid-West meeting at which the price of the dinner was $1.

Past-President Vincent urged that an aeronautic meeting be held by some Section in connection with a meeting of the Council. He spoke of the current popular misapprehension in regard to aeronautic affairs and thought that this could be dissipated by proper discussion in all Sections; the technical nature of the papers to vary according to the character of the audience which might be expected at different places. In connection with possible subjects for marine engineering papers, he suggested clutches and reverse gears and lubricating systems with reference to cooling the oil. B. B. Bachman, in speaking of Herbert Chase's paper on clutches, cited it as an example of what a paper of that type should contain. He suggested rear axles as a subject which could be similarly treated. He mentioned the possibility of the refinement of design in most parts of a car and the advisability of papers thereon, based upon a knowledge of past weaknesses and successes.

H. M. Crane urged that papers be prepared thoroughly, giving real facts and not be merely unimportant rambling discourses. He suggested lubricating systems and steering-gears as two subjects which might prove desirable for attention at Section meetings. L. S. Keilholtz said that the problems encountered in the design of isolated electric plants are numerous and not the same as those affecting car engines. He thought that a paper should be prepared in this connection. A. W. Scarratt, past-chairman, Minneapolis Section, talked on the impossibility of differentiating fuel and lubricating oil values from refiners' claims and said that experimentation must be conducted by machine manufacturers to enable the user to be informed as to the best grades to use. He urged that the tractor be adapted to industry as well as to the farm.

E. A. Johnston approved of the course of the Mid-West Section in its plans for the coming year. In suggesting subjects for meetings he spoke of the difficulty of uniform distribution of kerosene to all cylinders, the gas-producer with charcoal, and a comparison of steam and gas tractor efficiencies at various loads. J. H. Hunt, vice-chairman, Dayton Section, was confident that the program for the ensuing year could be laid out early in July except for those meetings which are to be held in conjunction with the Dayton Engineer's Club. T. C. Menges, vice-president of the Society, believed that more time could be given advantageously to stationary engines.

C. F. Scott, chairman of the Society Meetings Committee, stated that Section papers should not be secondary to those presented at Society meetings as to either subject or treatment. Good papers covering subjects adequately are necessary and if such papers are given, the attendance will take care of itself in Mr. Scott's opinion. He further believes that our engineering outlook should be years ahead instead of months.

A. K. Brumbaugh, past-chairman, Pennsylvania Section, said that in Philadelphia it is advisable to have semi-technical meetings because of the diversity of interest in that city. George E. Goddard, vice-chairman, Detroit Section, gave a number of specific suggestions for papers, including the design of pistons, methods of surfacing cylinder walls, and the insurance problem.

A number of other suggestions indicating the views of members as to specific subjects for Section papers are being received and further comment from members of the Society will be gladly considered.

Those who attended the luncheon were:

B. B. Bachman	J. H. Hunt
David Beecroft	A. E. Jackman
Geo. T. Briggs	E. A. Johnston
Horace A. Brown, Jr.	L. S. Keilholtz
A. K. Brumbaugh	C. D. LeFevre
R. S. Burnett	H. G. McComb
C. F. Clarkson	T. C. Menges
Hugh R. Corse	James A. Moyer
H. M. Crane	R. E. Northway
T. F. Cullen	B. S. Pfeiffer
Dr. H. C. Dickinson	A. W. Scarratt
G. W. Gilmer, Jr.	C. F. Scott
G. E. Goddard	H. W. Slauson
L. C. Hill	Lon R. Smith
K. K. Hoagg	C. B. Veal
H. L. Horning	J. G. Vincent

Applicants *Qualified*

The following applicants have qualified for admission to the Society between May 10 and June 10, 1921. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff) Affiliate; (S M) Service Member; (F M) Foreign Member; (E S) Enrolled Student.

ARDERN, JAMES S. (A) service manager, Northway Motors Corporation, *Boston*, (mail) 120 Peterboro Street.

ARMINGTON, A. P. (M) superintendent and tractor engineer, Euclid Crane & Hoist Co., *Euclid, Ohio*.

BASKERVILLE, DEAN E. (M) automotive body engineer, Dodge Bros., *Detroit*, (mail) 1455 Field Avenue.

BEST, NORMAN A. (M) engineer, metal division, Dow Chemical Co., *Midland, Mich.*, (mail) 717 Midland Avenue.

BOURNE, PHILLIPS P. (M) chief engineer, Blake Knowles works, Worthington Pump & Machinery Corporation, 265 Third Street, *East Cambridge, Mass.*

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Applicants *for* Membership

The applications for membership received between May 26 and June 22, 1921, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

ALLAN, ROBERT K., chief draftsman, Shefko Ball Bearing Co., Ltd., *Luton, Bedfordshire, England.*
ANDERSON, RALPH W., student, Buffalo Technical High School, *Buffalo.*
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BIRMINGHAM, C. J., service manager, Locomobile co., *Bridgeport, Conn.*
BRADY, GROVER C., battery repairman, Thompson Auto Co., *Windsor, Ont., Canada.*
BRUNELL, HOMER A., garage owner, B. & H. Garage, *Los Angeles, Cal.*
CARLE, A. E., engineer, American Bronze Corporation, *Berwyn, Pa.*
CARMAN, ARTHUR G., chemist and metallurgist, Franklin Die-Casting Corporation, *Syracuse, N. Y.*
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CLARK, RALPH R., student, Michigan Agricultural College, *East Lansing, Mich.*
COLBURN, HERBERT C., mechanical engineer, *Stockton, Cal.*
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DOTEN, EVERETT F., mechanical engineer, Muskegon Motor Specialties Co., *Muskegon, Mich.*
DUFFECK, FRANK W., general manager, Electric Power Maintenance Co., *Toledo.*
EDWARDS, WILLIAM H., research department, American Telephone & Telegraph Co., *New York City.*
FAGAN, XEN, general manager, Diamond branch of Rome Wire Co., *Buffalo.*
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